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Yano et al.

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(54) **VARIABLE VALVE OPERATING APPARATUS FOR INTERNAL COMBUSTION ENGINE**

(75) Inventors: **Toshiyuki Yano**, Gotemba (JP); **Shuichi Ezaki**, Susono (JP)

(73) Assignee: **TOYOTA JIDOSHA KABUSHIKI KAISHA**, Toyota-shi (JP)

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This patent is subject to a terminal disclaimer.

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USPC 123/90.15, 90.17, 90.16, 90.44; 464/160
See application file for complete search history.

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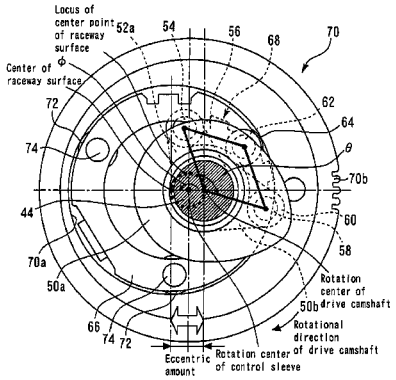
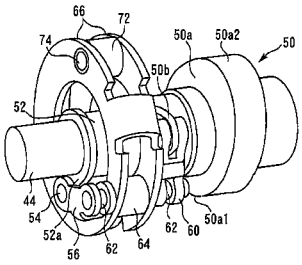
Primary Examiner — Ching Chang
(74) *Attorney, Agent, or Firm* — Oblon, McClelland, Maier & Neustadt, L.L.P.

(57) **ABSTRACT**

A variable valve operating apparatus for an internal combustion engine includes a drive camshaft, and a driven cam lobe that is rotatably supported by the drive camshaft. The variable valve operating apparatus further includes a control sleeve that has a raceway surface, a center of which is eccentric with respect to a center of rotation of its own. The variable valve operating apparatus further includes a link mechanism that is connected to each of the drive camshaft and the driven cam lobe and has a control roller which is in contact with the raceway surface. The variable valve operating apparatus further includes an actuator that drives the control sleeve. The variable valve operating apparatus further includes a control amount of the actuator is controlled to change a moving amount of the raceway surface in the above described plane direction in accordance with an operation condition of an internal combustion engine.

8 Claims, 12 Drawing Sheets





(51) **Int. Cl.**

F01L 1/356	(2006.01)
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F01L 1/047	(2006.01)
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USPC **123/90.16**; 123/90.15; 123/90.17

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Fig. 1

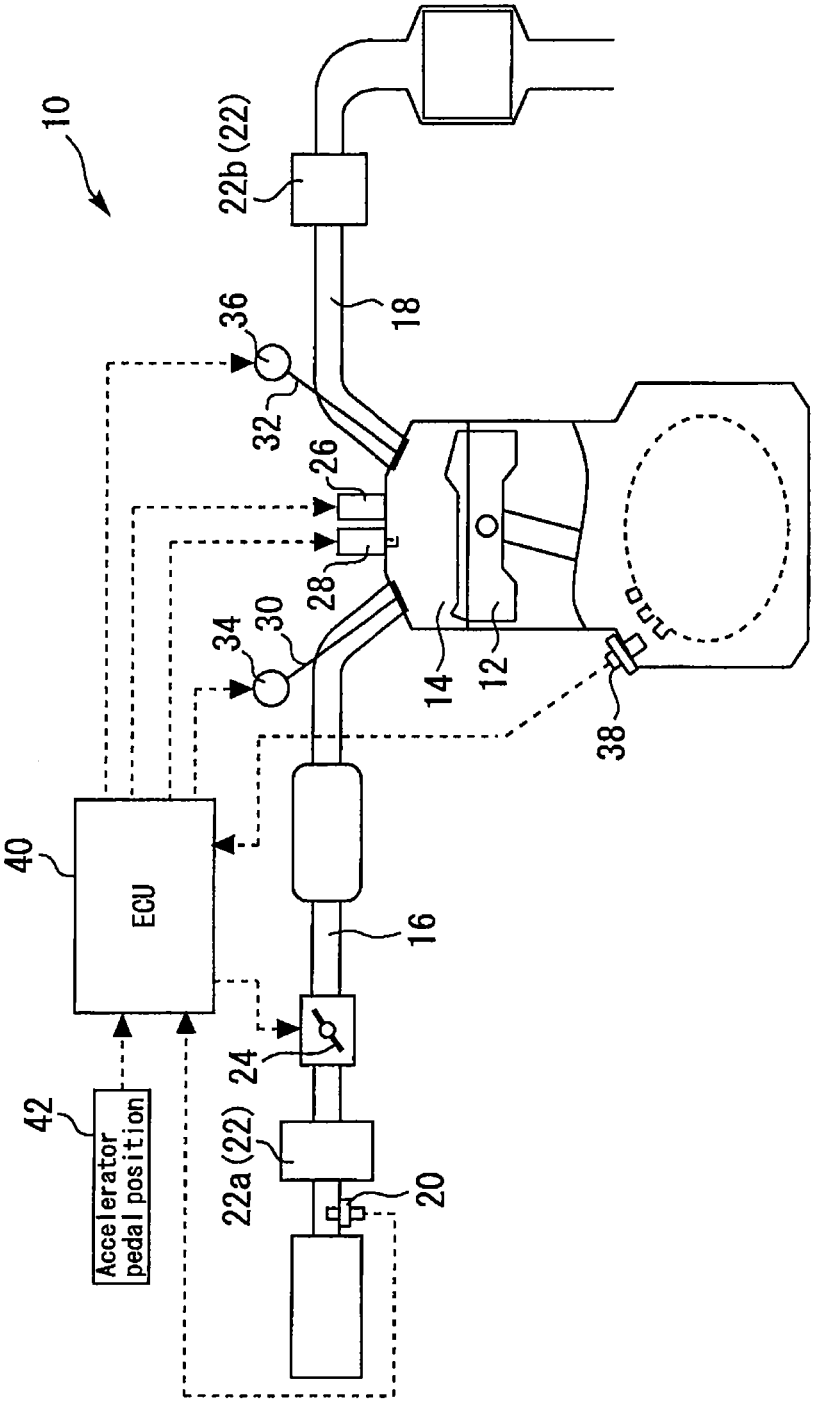

10
22b (22)
18
36
32
26
28
30
34
14
12
38
40
ECU
16
24
42
Accelerator pedal position
22a (22)
20

Fig. 2

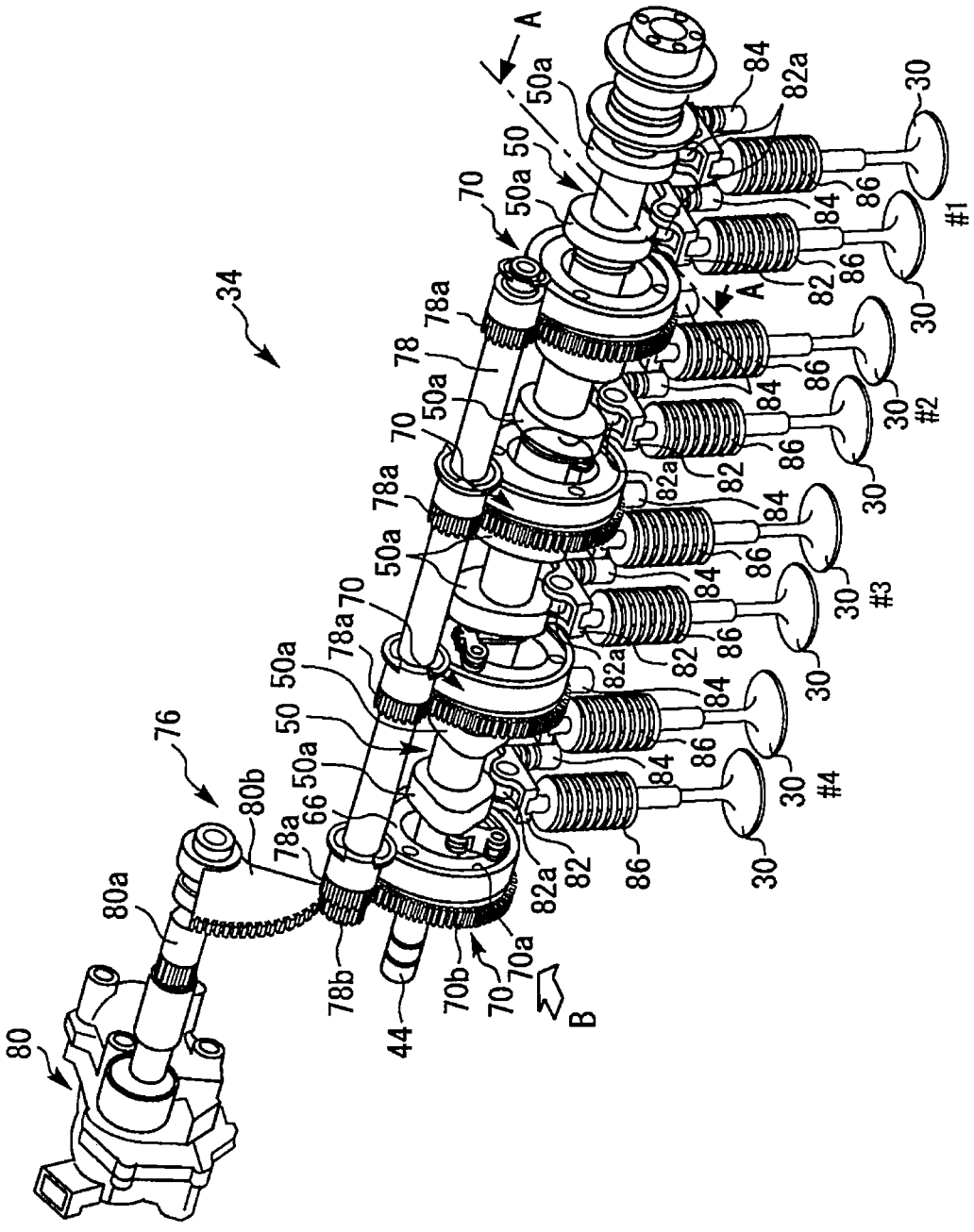

34
80
80a
80b
76
66
78
78a
78b
70
70a
70b
50
50a
82
82a
84
86
30
#1
#2
#3
#4
44
A
B

Fig. 3

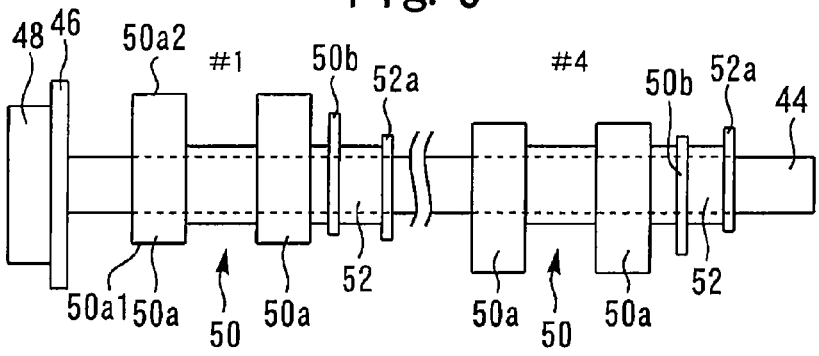

48
46
50a2
#1
50b
52a
#4
50b
52a
44
50a1
50a
50
50a
52
50a
50
50a
52

Fig. 4

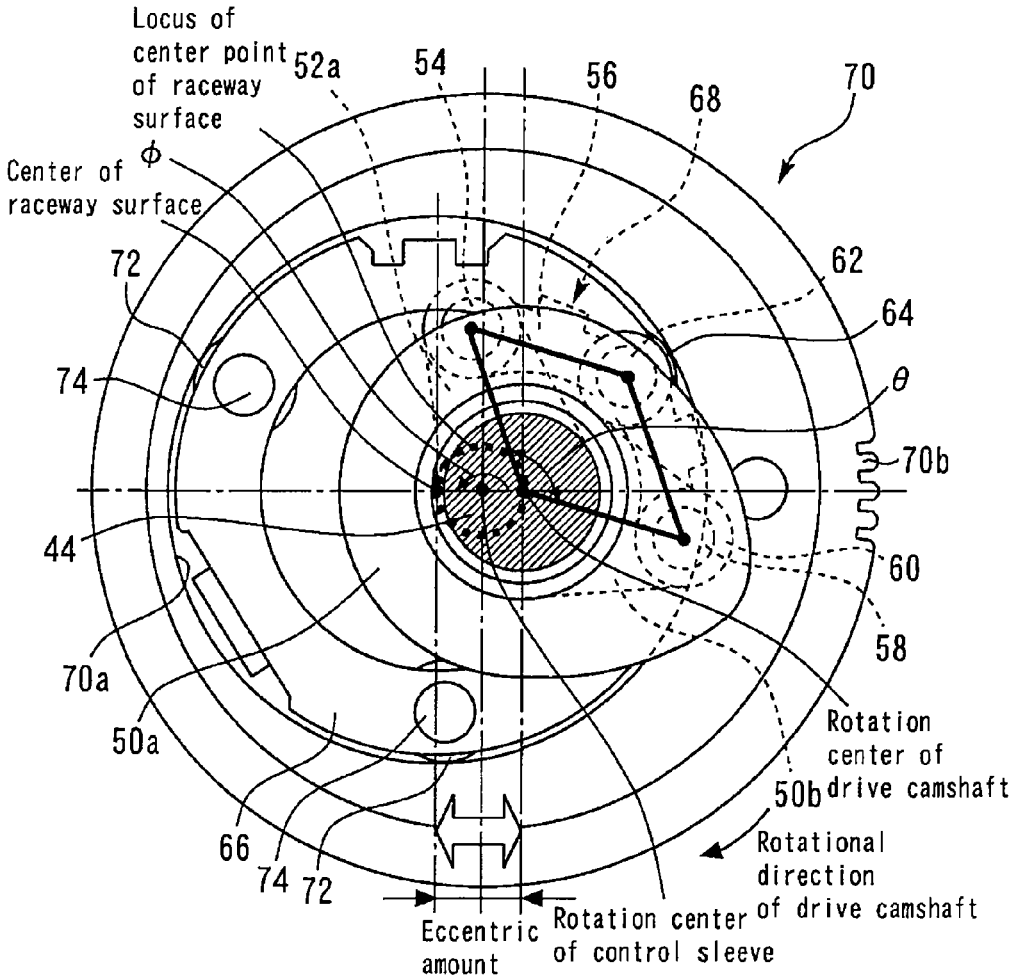

Locus of center point of raceway surface
Center of raceway surface
Φ
52a
54
56
68
70
72
62
64
74
θ
70b
44
60
58
70a
50a
Rotation center of drive camshaft
50b
Rotational direction of drive camshaft
66
74
72
Eccentric amount
Rotation center of control sleeve

Fig. 5

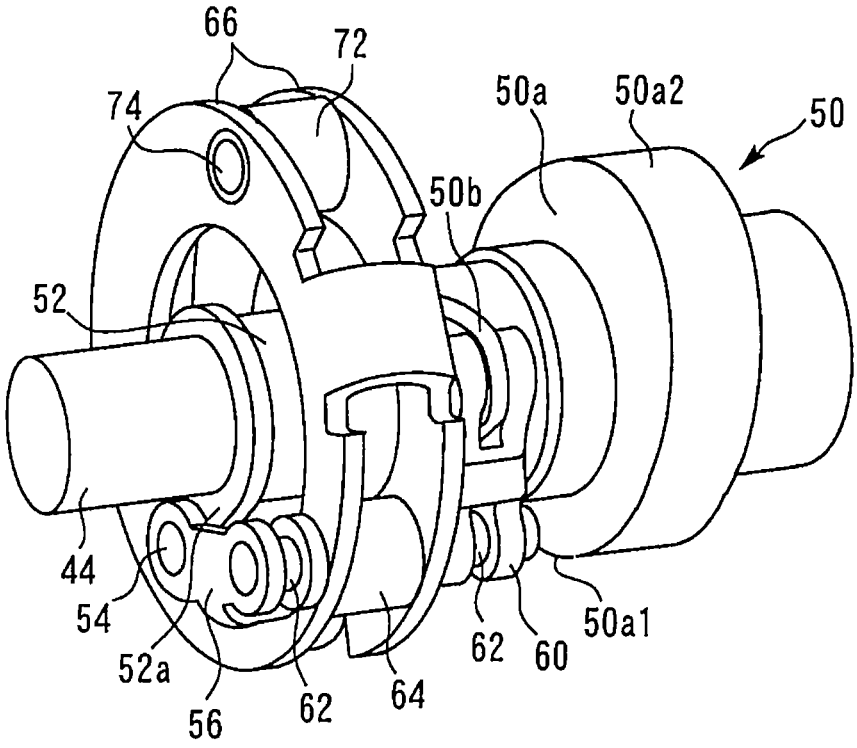


Fig. 6

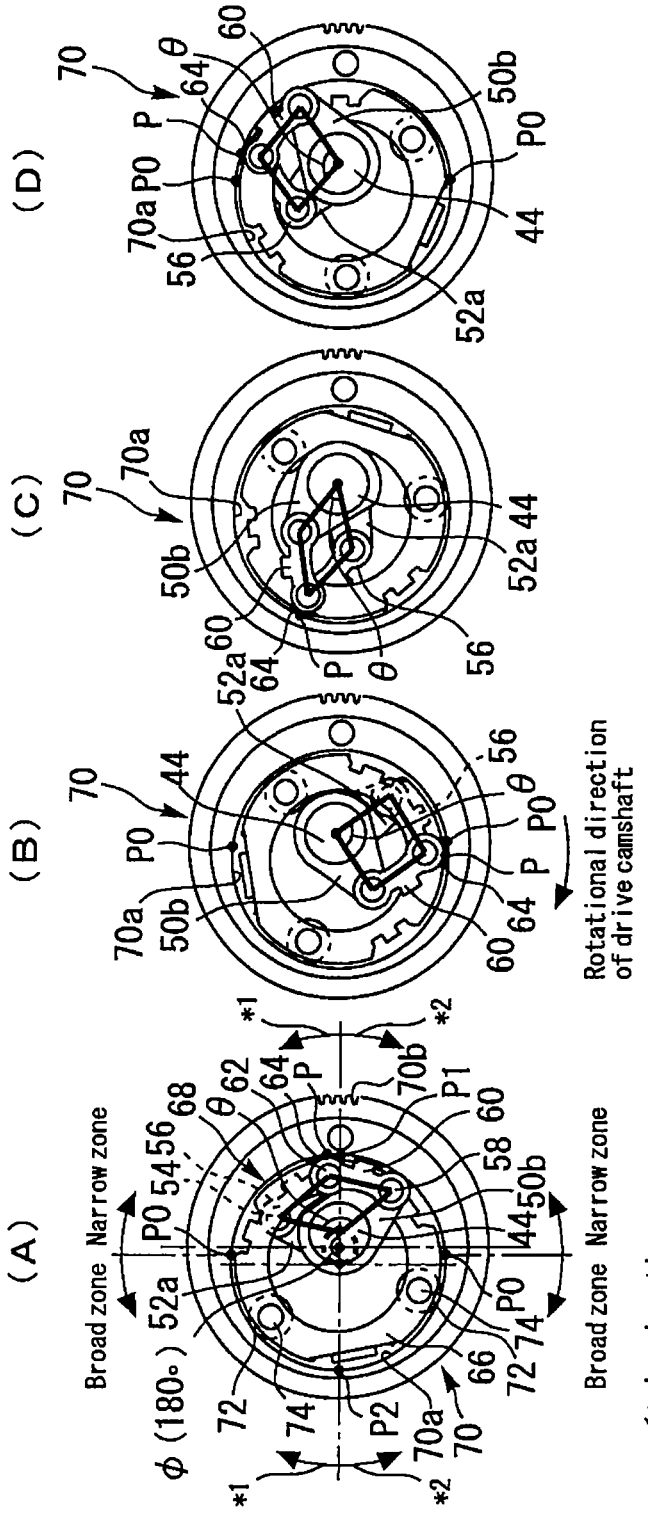

(A)
(B)
(C)
(D)
Broad zone
Narrow zone
Broad zone
Narrow zone
Rotational direction of drive camshaft
*1: Acceleration zone
*2: Deceleration zone

Fig. 7

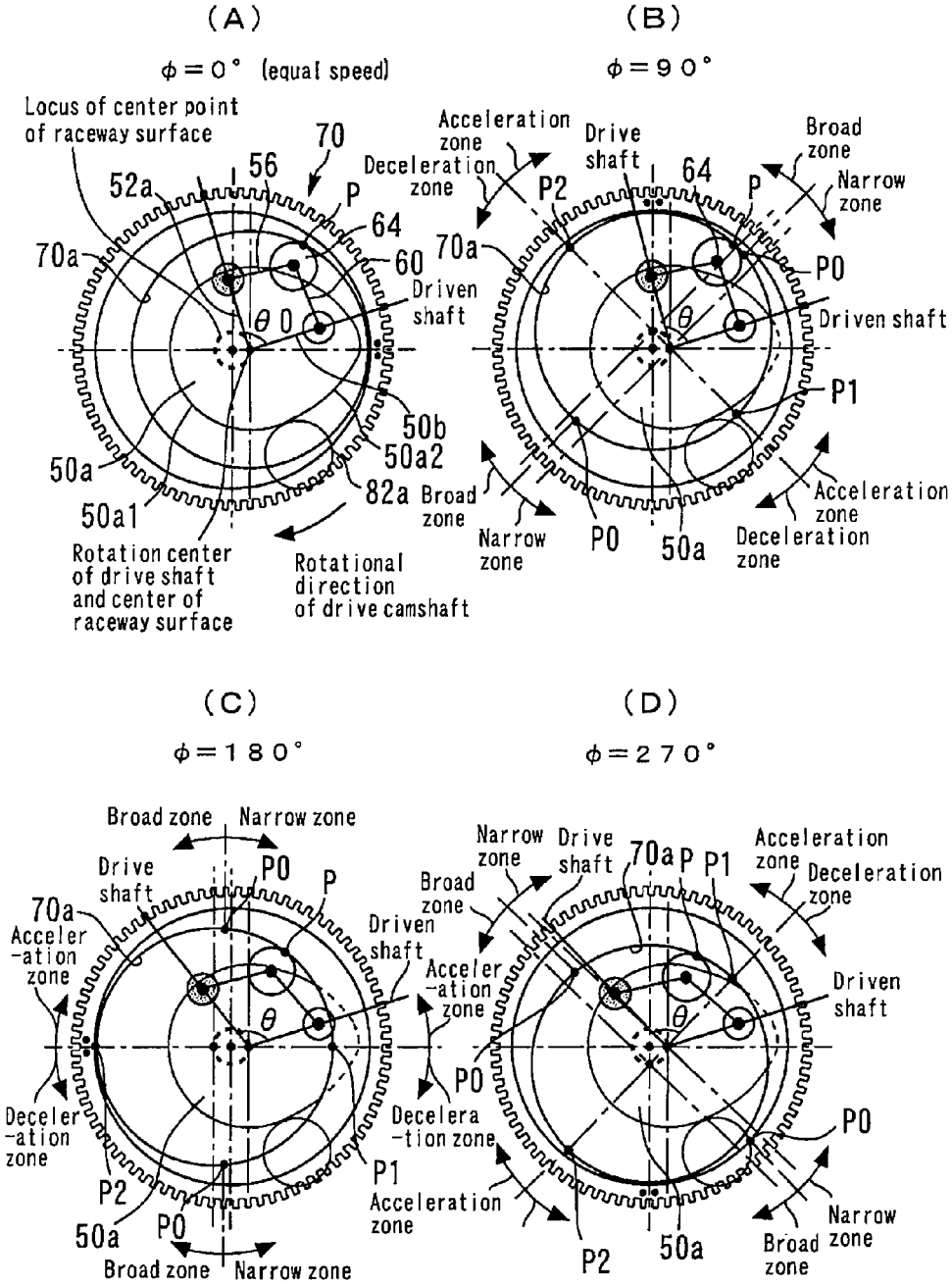

(A)
φ = 0° (equal speed)
Locus of center point of raceway surface
70
56
52a
P
64
60
70a
Driven shaft
θ 0
50b
50a2
50a
82a
50a1
Broad zone
Narrow zone
Rotation center of drive shaft and center of raceway surface
Rotational direction of drive camshaft
(B)
φ = 90°
Acceleration zone
Drive shaft
Deceleration zone
P2
64
P
Broad zone
Narrow zone
70a
P0
θ
Driven shaft
P1
Acceleration zone
Deceleration zone
Narrow zone
P0
50a
(C)
φ = 180°
Broad zone
Narrow zone
Drive shaft
P0
P
70a
Acceler-ation zone
Driven shaft
θ
Deceler-ation zone
P2
50a
P0
P1
Acceleration zone
Decelera-tion zone
Broad zone
Narrow zone
(D)
φ = 270°
Narrow zone
Drive shaft
70a
P
P1
Acceleration zone
Deceleration zone
Broad zone
Acceler-ation zone
Driven shaft
θ
P0
P0
P2
50a
Narrow zone
Broad zone

Fig. 8

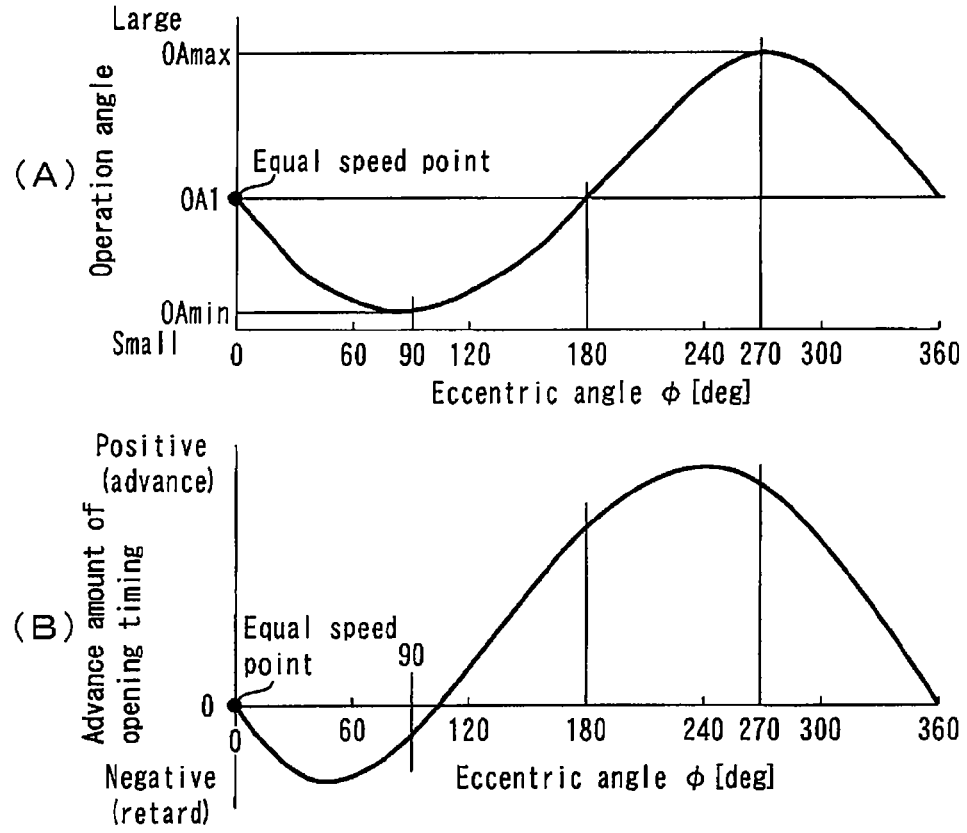

(A)
Large
OAmax
Operation angle
Equal speed point
OA1
OAmin
Small
0
60
90
120
180
240
270
300
360
Eccentric angle φ [deg]
(B)
Positive (advance)
Advance amount of opening timing
Equal speed point
90
0
0
60
120
180
240
270
300
360
Negative (retard)
Eccentric angle φ [deg]

Fig. 9

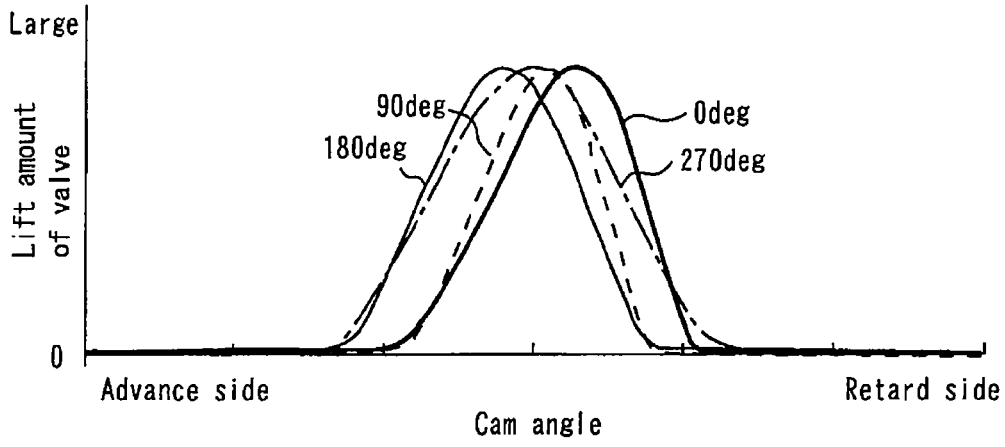

Large
Lift amount of valve
0
90deg
180deg
0deg
270deg
Advance side
Retard side
Cam angle

Fig. 10

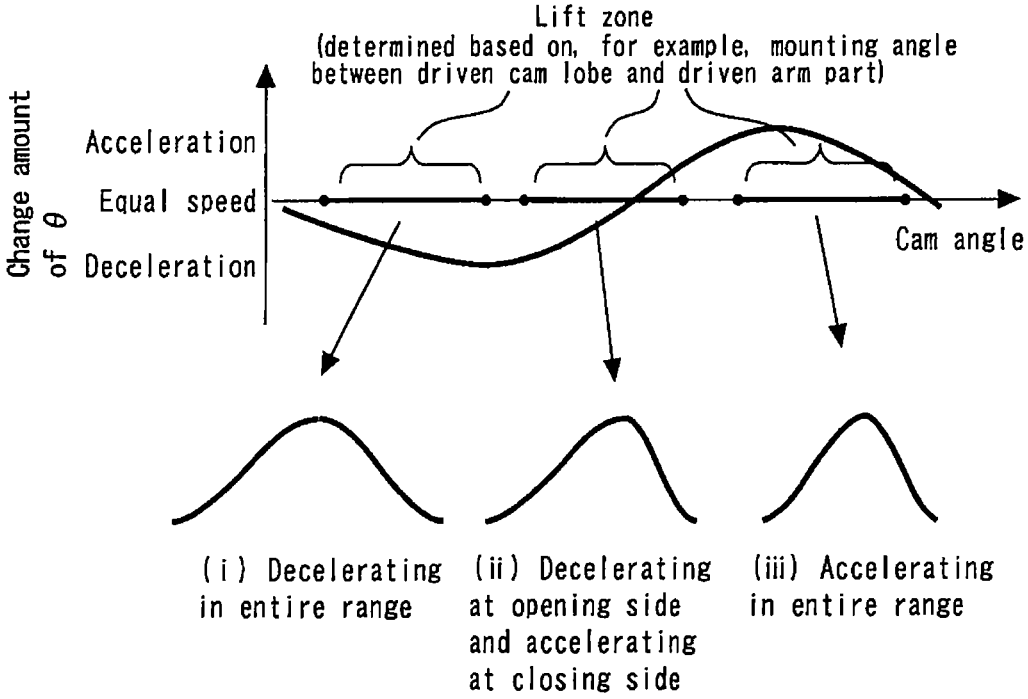

Lift zone
(determined based on, for example, mounting angle between driven cam lobe and driven arm part)
Change amount of θ
Acceleration
Equal speed
Deceleration
Cam angle
(i) Decelerating in entire range
(ii) Decelerating at opening side and accelerating at closing side
(iii) Accelerating in entire range

Fig. 11

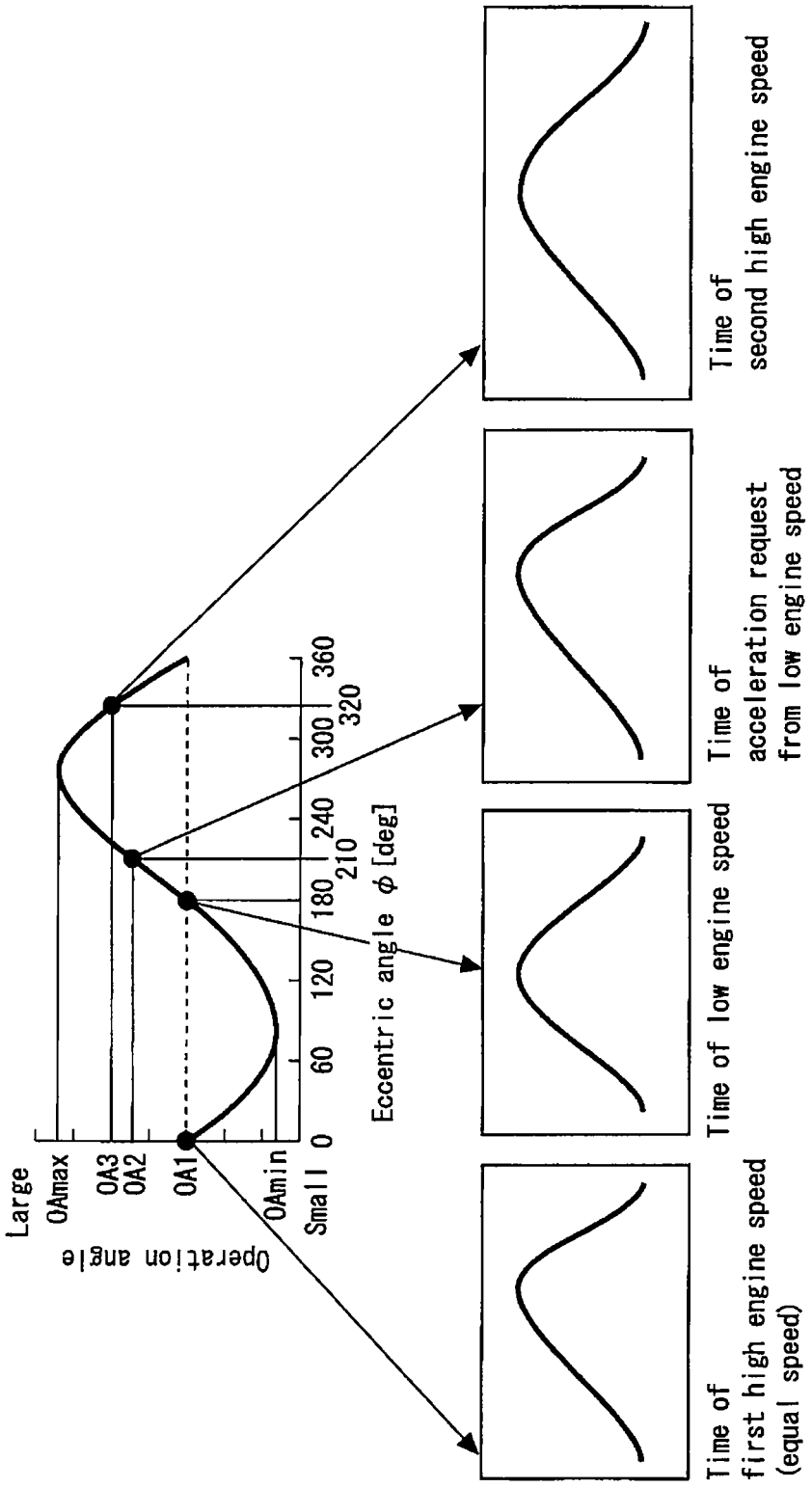

Large
OAmax
OA3
OA2
OA1
OAmin
Small
Operation angle
0
60
120
180
210
240
300
320
360
Eccentric angle φ [deg]
Time of first high engine speed (equal speed)
Time of low engine speed
Time of acceleration request from low engine speed
Time of second high engine speed

Fig. 12

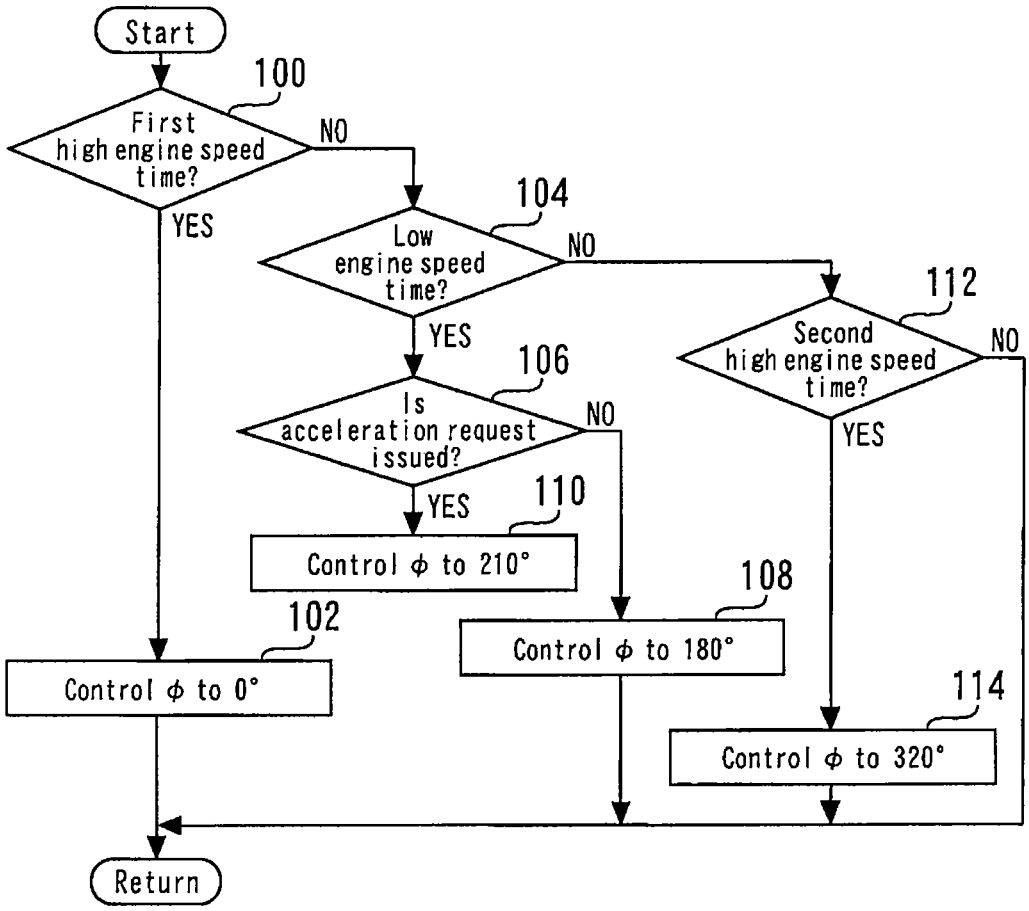

Start
100
First high engine speed time?
NO
YES
104
Low engine speed time?
NO
YES
106
Is acceleration request issued?
NO
YES
110
Control ϕ to 210°
112
Second high engine speed time?
NO
YES
102
Control ϕ to 0°
108
Control ϕ to 180°
114
Control ϕ to 320°
Return

Fig. 13

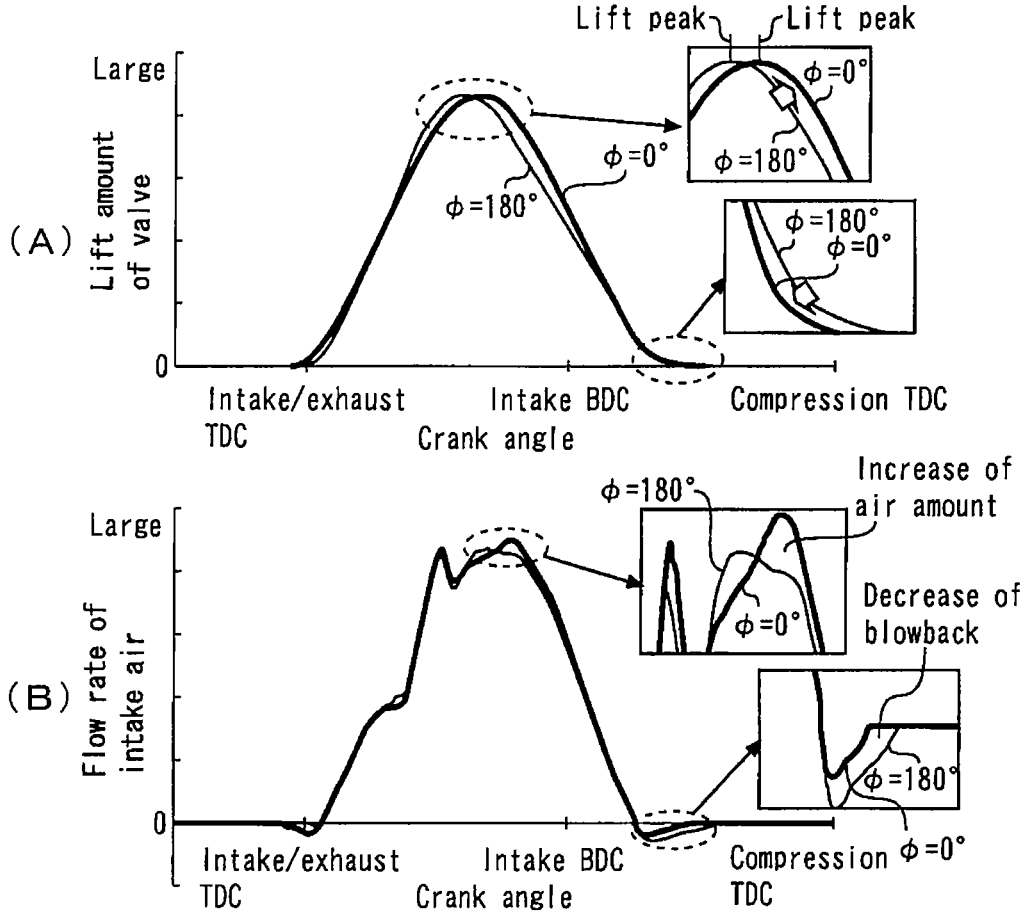

(A)
Lift amount of valve
Large
0
Lift peak
Lift peak
ϕ=0°
ϕ=180°
ϕ=0°
ϕ=180°
ϕ=180°
ϕ=0°
Intake/exhaust TDC
Intake BDC
Crank angle
Compression TDC
(B)
Flow rate of intake air
Large
0
ϕ=180°
Increase of air amount
ϕ=0°
Decrease of blowback
ϕ=180°
ϕ=0°
Intake/exhaust TDC
Intake BDC
Crank angle
Compression TDC

Fig. 14

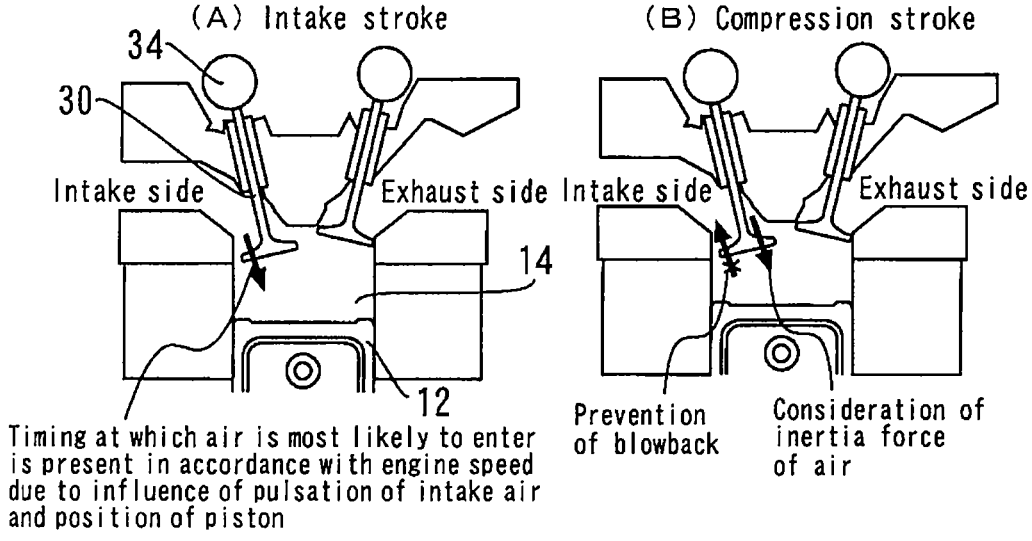

(A) Intake stroke
34
30
Intake side
Exhaust side
14
12
Timing at which air is most likely to enter is present in accordance with engine speed due to influence of pulsation of intake air and position of piston
(B) Compression stroke
Intake side
Exhaust side
Prevention of blowback
Consideration of inertia force of air

Fig. 15

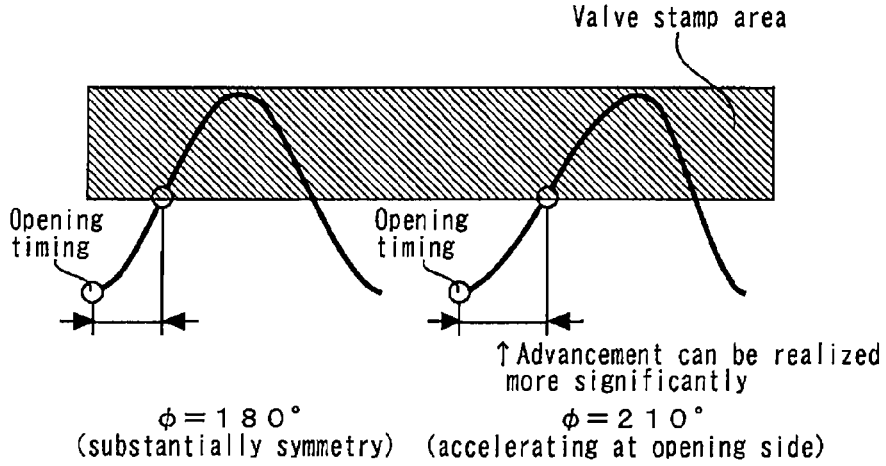
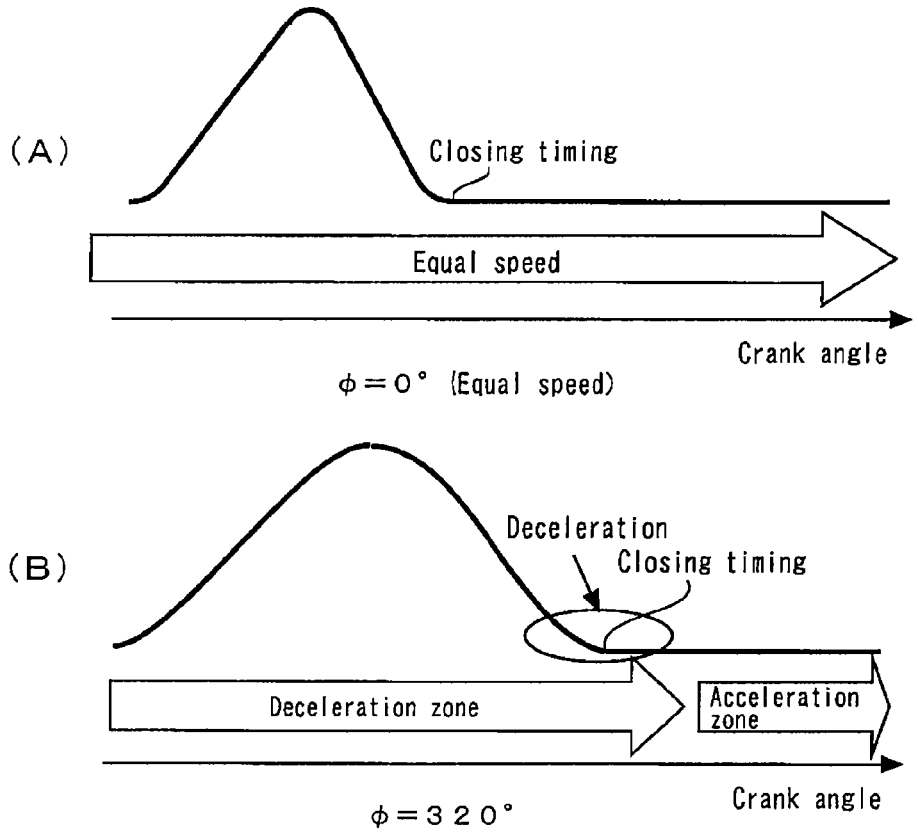

Valve stamp area
Opening timing
Opening timing
↑Advancement can be realized more significantly
φ＝１８０°
(substantially symmetry)
φ＝２１０°
(accelerating at opening side)

Fig. 16


(A)
Closing timing
Equal speed
Crank angle
φ＝０° (Equal speed)
(B)
Deceleration
Closing timing
Deceleration zone
Acceleration zone
Crank angle
φ＝３２０°

VARIABLE VALVE OPERATING APPARATUS FOR INTERNAL COMBUSTION ENGINE

TECHNICAL FIELD

The present invention relates to a variable valve operating apparatus for an internal combustion engine, and particularly relates to a variable valve operating apparatus for an internal combustion engine which is capable of changing a rotational speed of a driven cam lobe for an interval in which a drive camshaft makes one rotation.

BACKGROUND ART

Conventionally, for example, Patent Document 1 discloses a valve operating apparatus for an internal combustion engine including a configuration in which an electric motor rotationally drives a drive camshaft to which a driven cam lobe which drives a valve is fixed. Further, the conventional valve operating apparatus includes a motor control device which controls the rotational speed of the electric motor. According to the configuration like this, the rotational speed of the electric motor is changed by the motor control device, and thereby the rotational speed of the driven cam lobe for an interval in which the drive camshaft makes one rotation can be increased and decreased.

The applicant recognizes the documents described as follows including the above described document, as the documents relating to the present invention.

CITATION LIST

Patent Documents

Patent Document 1: Japanese Laid-open Patent Application Publication No. 2005-180238

Patent Document 2: Japanese Laid-open Patent Application Publication No. 2009-57868

SUMMARY OF INVENTION

Technical Problem

In the variable valve operating apparatus that adopts an ordinary configuration which drives the drive camshaft by the rotational force of the crankshaft which is transmitted via a timing chain or timing belt, unlike the configuration of the conventional valve operating apparatus described above, there is also a demand to realize the function of making it possible to change the rotational speed of the driven cam lobe for an interval in which the drive camshaft makes one rotation. In order to enhance various performances (fuel consumption, output power, exhaust emission and the like) of the internal combustion engine which includes a variable valve operating apparatus having the above described function, it is desirable to perform control which takes advantage of the above described function.

The present invention is made to solve the problem as described above, and an object of the invention is to provide a variable valve operating apparatus for an internal combustion engine which can favorably enhance various performances of the internal combustion engine while realizing the function which can change a rotational speed of an driven cam lobe for an interval in which a drive camshaft makes one rotation, in a case of using a configuration which drives the drive camshaft by a rotational force of a crankshaft.

Solution to Problem

The present invention is a variable valve operating apparatus for an internal combustion engine, and includes a drive camshaft, a driven cam lobe, a guide member, a link mechanism, contact maintaining means, an actuator and control means.

The drive camshaft is rotationally driven by a rotational force of a crankshaft. The driven cam lobe is concentric with the drive camshaft, and is rotatably supported by the drive camshaft. The guide member has a raceway surface formed so as to surround the drive camshaft. The link mechanism is connected to each of the drive camshaft and the driven cam lobe, has a contact member which is in contact with the raceway surface, and changes a rotation angle of the driven cam lobe with respect to the drive camshaft in association with a positional change of the contact member with respect to a center of rotation of the drive camshaft. The contact maintaining means maintains contact between the contact member that revolves around the drive camshaft and the raceway surface, while the drive camshaft makes one rotation. The actuator moves the raceway surface in a plane direction orthogonal to an axial line of the drive camshaft. The control means controls a control amount of the actuator to change a moving amount of the raceway surface in the plane direction, in accordance with an operation condition of the internal combustion engine.

According to the present invention, the raceway surface is moved in the plane direction orthogonal to the axial line of the drive camshaft by the actuator, whereby the position of the raceway surface on the above described plane changes, and the positional change of the contact member of the link mechanism with respect to the center of rotation of the drive camshaft occurs. With this, the relative rotation angle of the driven cam lobe with respect to the drive camshaft for an interval in which the drive camshaft makes one rotation changes. As a result, the rotational speed of the driven cam lobe for the interval in which the drive camshaft makes one rotation changes in accordance with the control position of the raceway surface of the guide member. The present invention includes the control means that controls the control amount of the actuator to change the moving amount of the raceway surface in the above described plane direction in accordance with the operation condition of the internal combustion engine. By this, the relation between each operation condition and the control amount of the actuator is set in advance to obtain the characteristic (shape) of the desired lift curve in each operation condition, whereby various performances of the internal combustion engine can be favorably enhanced while the function of being able to change the rotational speed of the driven cam lobe in the interval in which the drive camshaft makes one rotation is realized in the case of using the configuration which drives the drive camshaft by the rotational force of the crankshaft.

Further, the actuator in the present invention may rotationally drive the guide member. The raceway surface may be a circumferential surface, and may be included by the guide member in a state in which a center of the raceway surface is eccentric with respect to a center of rotation of the guide member. The control amount of the actuator by the control means may be a rotation angle of the guide member.

According to the configuration as above, the variable valve operating apparatus can be realized, which can change the rotational speed of the driven cam lobe in the interval in which the drive camshaft makes one rotation by controlling the rotation angle of the guide member, by using the actuator having the configuration which moves the raceway surface

having the center eccentric with respect to the center of rotation of the guide member in the above described plane direction by rotationally driving the guide member. According to the configuration as described above, the first and the second lift curves with the same or substantially the same operation angle values obtained, and with timings in which the lift amounts show peaks differing from each other are obtained when the rotation angle of the guide member which is the control amount of the actuator is changed.

Further, the control means in the present invention may include at least a first rotation angle and a second rotation angle as target values of the rotation angle of the guide member. The first rotation angle may be a rotation angle of the guide member at a time when a predetermined operation angle value for an operation angle of a valve which is driven by the driven cam lobe is obtained. The second rotation angle may be a rotation angle of the guide member at a time when a same or substantially the same operation angle value as the operation angle value which is obtained at a time of control to the first rotation angle is obtained, and when a second lift curve with timing in which a lift amount shows a peak differing as compared with a first lift curve of the valve which is obtained at the time of the control to the first rotation angle is obtained. The control means may discriminatingly use the first rotation angle and the second rotation angle under at least two kinds of operation conditions which differ in engine speed from each other.

By this, as compared with the conventional variable valve operating apparatus which can obtain only the lift curve with one characteristic (one shape) for one operation angle, control of the valve opening characteristic of the valve which can better satisfy demands in respective operation conditions can be performed by discriminatingly using, in accordance with the operation conditions, the first and the second lift curves with which the same or substantially the same operation angle values are obtained and the timings in which the lift amounts showing peaks differing.

Further, the valve in the present invention may be an intake valve. The second lift curve may be set so that timing in which the lift amount shows the peak is delayed as compared with the first lift curve. The control means may use the second rotation angle in an operation condition with a higher engine speed than an operation condition for which the first rotation angle is used.

The timing in which an amount of air easily enters the cylinder during the lift zone changes in accordance with the engine speed. More specifically, as the engine speed becomes higher, the above described timing becomes relatively later. According to the configuration as described above, a high lift amount of the intake valve can be ensured in the timing in which much air easily enters during the lift zone, irrespective of the value of the engine speed, and therefore, the amount of the air filled in the cylinder can be increased. Therefore, the output performance of the internal combustion engine can be enhanced.

Further, the second lift curve in the present invention may be set so that a lift amount in a predetermined section in a vicinity of a closing timing of the valve becomes small, as compared with the first lift curve.

By this, the decrease of the amount of the air filled in the cylinder due to the influence of blowback of the intake air can be prevented, at the time of use of the second lift curve.

Further, a valve which is driven by the driven cam lobe in the present invention may be an intake valve. When an acceleration request is issued at a time of a low engine speed, the control means may control the control amount of the actuator so that a relative rotational speed of the driven cam lobe with respect to the drive camshaft increases in a predetermined section in a vicinity of an opening timing of the intake valve.

According to the configuration as above, when an acceleration request is issued at the time of the low engine speed, a clearance between the piston and the intake valve can be ensured more sufficiently in the predetermined section in the vicinity of the opening timing of the intake valve. As a result, the opening timing of the intake valve is advanced significantly, and the valve overlap amount with the exhaust valve can be effectively increased. By this, in the state in which the intake pressure becomes higher than the exhaust pressure at the time of acceleration, the scavenging effect is enhanced by the increase of the valve overlap amount. As a result, the output performance of the internal combustion engine can be enhanced. Further, under the situation in which the intake pressure is lower than the exhaust pressure at the time of acceleration, the internal EGR gas amount increases by enlargement of the valve overlap amount. As a result, the fuel consumption performance and the exhaust emission performance of the internal combustion engine can be enhanced.

Further, when an engine speed is higher than a predetermined speed, the control means in the present invention may control the control amount of the actuator so that a relative rotational speed of the driven cam lobe with respect to the drive camshaft decreases in a predetermined section in a vicinity of a closing timing of a valve which is driven by the driven cam lobe.

By this, even under the situation in which the engine speed exceeds the normal use range for some reason, occurrence of an abnormal valve operation such as valve bounce can be avoided by reducing the rate of acceleration at the time of seating of the valve. Therefore, reliability of the variable valve operating apparatus can be enhanced.

BRIEF DESCRIPTION OF DRAWINGS

FIG. **1** is a view for explaining a system configuration of an internal combustion engine equipped with variable valve apparatuses of the present invention;

FIG. **2** is a perspective view schematically showing an entire configuration of the variable intake valve operating apparatus shown in FIG. **1**;

FIG. **3** is a view for explaining a configuration around a drive camshaft included by the variable intake valve operating apparatus shown in FIG. **1**;

FIG. **4** is a sectional view of the variable intake valve operating apparatus cut along the A-A-line shown in FIG. **2**;

FIG. **5** is a perspective view of a configuration around the drive camshaft as seen from the arrow B direction in FIG. **2**;

FIG. **6** is a view expressing an operation of a link mechanism for an interval in which the drive camshaft makes one rotation (mainly a change of the rotation angle θ during one rotation of the drive camshaft) as an example;

FIG. **7** is a schematic view for explaining the operations of the variable intake valve operating apparatus when the eccentric angle ϕ is changed by 90° at a time;

FIG. **8** is a diagram expressing a tendency of a change of valve opening characteristics of an intake valve in association with a change of the eccentric angle ϕ in the variable intake valve operating apparatus;

FIG. **9** is a diagram expressing each lift curve of the intake valve which is obtained when the eccentric angle ϕ is changed by 90° at a time;

FIG. **10** is a diagram expressing a relation between the lift zone of the intake valve and the acceleration/deceleration zones;

FIG. **11** is a diagram expressing a relation between the eccentric angle ϕ and each lift curve of the intake valve which is to be changed in accordance with the operation condition of the internal combustion engine in embodiment 1 of the present invention;

FIG. **12** is a flowchart of a routine that is executed in embodiment 1 of the present invention;

FIG. **13** is a view comparing the lift curves of the intake valve and the flow rates of the intake air which is taken into the cylinder during the lift zone in the case of the eccentric angle ϕ being 0° and in the case of the eccentric angle ϕ being 180°;

FIG. **14** is a view for explaining a behavior of intake air in an intake stroke and a compression stroke;

FIG. **15** is a diagram which compares the two lift curves with the eccentric angles ϕ of 180° and 210°, in relation with a valve stamp area; and

FIG. **16** is a diagram which compares the lift curve with the eccentric angle ϕ of 320° with the lift curve (ϕ=0°) at the time of the equal speed.

DESCRIPTION OF EMBODIMENT

Embodiment 1

[System Configuration of an Internal Combustion Engine]

FIG. **1** is a view for explaining a system configuration of an internal combustion engine **10** equipped with variable valve apparatuses **34** and **36** of the present invention. Here, the internal combustion engine **10** is an in-line 4-cylinder type engine having four cylinders (#1 to #4) as an example.

A piston **12** is provided in a cylinder of the internal combustion engine **10**. In the cylinder of the internal combustion engine **10**, a combustion chamber **14** is formed atop the piston **12**. An intake passage **16** and an exhaust passage **18** communicate with the combustion chamber **14**. An air flow meter **20** which outputs a signal corresponding to a flow rate of air taken into the intake passage **16** is provided in the vicinity of an inlet of the intake passage **16**.

A compressor **22***a* of a turbo supercharger **22** is disposed in the intake passage **16** at a downstream side of the air flow meter **20**. Further, a turbine **22***b* of the turbo supercharger **22** is disposed in the exhaust passage **18**. An electronically controlled throttle valve **24** is provided in the intake passage **16** at a downstream side from the compressor **22***a*. Further, each of the cylinders of the internal combustion engine **10** is provided with a fuel injection valve **26** for directly injecting a fuel into the combustion chamber **14** (into the cylinder), and an ignition plug **28** for igniting a mixture gas.

An intake port and an exhaust port are respectively provided with an intake valve **30** and an exhaust valve **32** for bringing the combustion chamber **14** and the intake passage **16**, or the combustion chamber **14** and the exhaust passage **18** into a continuing state or a cutoff state. The intake valve **30** and the exhaust valve **32** are respectively driven by a variable intake valve operating apparatus **34** and a variable exhaust valve operating apparatus **36**. Detailed configurations of these variable valve apparatuses **34** and **36** and operations thereof will be described later with reference to FIG. **2** to FIG. **9**.

Further, the system shown in FIG. **1** includes an ECU (Electronic Control Unit) **40**. Various sensors for detecting an operation state of the internal combustion engine **10** such as a crank angle sensor **38** for detecting an engine speed, and an accelerator position sensor **42** for detecting a pedal position of an accelerator of a vehicle equipped with the internal combustion engine **10** in addition to the aforementioned air flow meter **20** are connected to an input section of the ECU **40**. Further, various actuators for controlling the operation of the internal combustion engine **10** such as the throttle valve **24**, the fuel injection valve **26**, the ignition plug **28** and the variable valve apparatuses **34** and **36** described above are connected to an output section of the ECU **40**. The ECU **40** controls the operation state of the internal combustion engine **10** by driving the above described various actuators in accordance with predetermined programs and the sensor outputs.

Next, with reference to FIG. **2** to FIG. **10**, configurations of the variable valve apparatuses **34** and **36** and operations thereof will be described in detail. Here, the variable intake valve operating apparatus **34** is described as an example, but the variable exhaust valve operating apparatus **36** is assumed to be basically configured similarly to the variable intake valve operating apparatus **34** except for a cam profile of a driven cam lobe and the like.

[Configuration of the Variable Valve Operating Apparatus]

FIG. **2** is a perspective view schematically showing an entire configuration of the variable intake valve operating apparatus **34** shown in FIG. **1**. FIG. **3** is a view for explaining a configuration around a drive camshaft **44** included by the variable intake valve operating apparatus **34** shown in FIG. **1**.

As shown in FIGS. **2** and **3**, the variable intake valve operating apparatus **34** includes the drive camshaft **44**. The drive camshaft **44** is connected to a crankshaft (not illustrated) via a timing pulley **46**, a timing chain and the like (not illustrated), and is configured to rotate at half speed of the crankshaft. As shown in FIG. **3**, a known variable valve timing (VVT) mechanism **48** which can change a rotation phase of the drive camshaft **44** with respect to the rotation of the crankshaft is interposed between the drive camshaft **44** and the timing pulley **46**. According to the VVT mechanism **48** like this, an opening timing and a closing timing (more specifically, an entire lift curve) can be changed to an advance side or a retard side in a crank angle base without changing an operation angle of the intake valve **30**.

As shown in FIG. **3**, on the drive camshaft **44**, a cam piece **50** is mounted for each cylinder. The cam piece **50** is concentric with the drive camshaft **44**, and is rotatably supported by the drive camshaft **44**. At the cam piece **50**, two driven cam lobes **50***a* for driving the intake valve **30** are formed. The driven cam lobe **50***a* includes a base circle part **50***a***1** in a circular arc shape coaxial with the drive camshaft **44**, and a nose part **50***a***2** which is formed by expanding a part of the base circle toward an outside in a radial direction.

Further, on the drive camshaft **44**, a drive arm **52** having a drive arm part **52***a* which projects to outside in the radial direction of the drive camshaft **44** is mounted for each cylinder. The drive arm **52** is integrally fixed to the drive camshaft **44** by using a predetermined fixing member (not illustrated). Further, a driven arm part **50***b* which projects to outside in the radial direction of the drive camshaft **44** is integrally formed at the cam piece **50** in the vicinity of the driven cam lobe **50***a* which is closer to the drive arm **52** for the same cylinder.

The description will be continued with FIG. **4** and FIG. **5** newly added.

FIG. **4** is a sectional view of the variable intake valve operating apparatus **34** cut along the A-A-line shown in FIG. **2**. FIG. **5** is a perspective view of a configuration around the drive camshaft **44** as seen from the arrow B direction in FIG. **2**. In FIG. **5**, illustration of a control sleeve **70** is omitted.

As shown in FIGS. **4** and **5**, one end of a drive link **56** is rotatably connected to the drive arm part **52***a* via a camshaft side rotary shaft **54**. Further, one end of a driven link **60** is rotatably connected to the driven arm part **50***b* via a cam lobe side rotary shaft **58**. Further, the other end of the drive link **56** and the other end of the driven link **60** are connected via a

control roller side rotary shaft **62**. A control roller **64** and a link plate **66** are interposed in a region between the drive link **56** and the driven link **60** on the control roller side rotary shaft **62**.

As above, the variable intake valve operating apparatus **34** of the present embodiment includes a link mechanism **68** which is a four joint link connected into a pantograph shape (rhombus shape) as shown in FIG. **4** by the drive arm part **52***a* and the driven arm part **50***b* with an axial center of the drive camshaft **44** as a common center of rotation, and the drive link **56** and the driven link **60**. Further, as shown in FIG. **4**, in the present embodiment, the driven link **60** is disposed at a front side in the rotational direction of the drive camshaft **44** with respect to the drive link **56** in a state in which the control roller **64** is interposed between the driven link **60** and the drive link **56**.

The link plate **66** is formed by two plate parts formed into ring shapes being folded to be concentric with each other as shown in FIG. **5**. In a state in which the drive camshaft **44** is penetrated through an inside thereof, and the link plate **66** sandwiches the control roller **64** from outside, the link plate **66** is disposed on the control roller side rotary shaft **62**.

As shown in FIG. **4**, a raceway surface **70***a* of the control sleeve **70** is disposed at an outer circumferential side of the link plate **66** in such a manner as to surround the link plate **66** the inside of which is penetrated through by the drive camshaft **44**. More specifically, the raceway surface **70***a* of the present embodiment is configured by a circumferential surface. Further, the above described control roller **64** is rotatably supported by the control roller side rotary shaft **62** at a position where the control roller **64** is in contact with the raceway surface **70***a* so as to be able to roll on the raceway surface **70***a* in conjunction with the rotation of the drive camshaft **44**.

Further, as shown in FIG. **4**, inside the link plate **66**, two holding rollers **72** are rotatably mounted in positions where the two holding rollers **72** are in contact with the raceway surface **70***a* via holding rotary shafts **74**, besides the control roller **64**. More specifically, the three rollers **64** and **72** are mounted to the link plate **66** so that disposition of the three rollers **64** and **72** including the two holding rollers **72** in addition to the control roller **64** is at equiangular spaces with the drive camshaft **44** as a center. According to the configuration like this, the link plate **66** rotates inside the raceway surface **70***a* while the control roller **64** and the two holding rollers **72** roll on the raceway surface **70***a*, with rotation of the drive camshaft **44**. More specifically, the link plate **66** has the position in the radial direction of the drive camshaft **44** defined by the raceway surface **70***a* via the control roller **64** and the holding rollers **72**, and the position on the raceway surface **70***a*, of the control roller **64** which is mounted to the link plate **66** is defined. Therefore, the control roller **64** rolls on the raceway surface **70***a* in a state in which the control roller **64** is always in contact with the raceway surface **70***a* with rotation of the drive camshaft **44**. As a result that the position of the control roller **64** is defined, a relative rotation angle θ of the driven cam lobe **50***a* with respect to the drive camshaft **44** is also determined via the drive link **56** and the driven link **60**.

(Definition of the Rotation Angle θ)

In this case, the above described rotation angle θ is defined as an angle formed by a straight line (drive axis) connecting a center point of the drive camshaft **44** and a center point of the camshaft side rotary shaft **54**, and a straight line (driven axis) connecting the center point of the drive camshaft **44** and a center of the cam lobe side rotary shaft **58**, as seen from an axial direction of the drive camshaft **44**, as shown in FIG. **4**.

Further, the variable intake valve operating apparatus **34** of the present embodiment includes an actuator **76** for rotationally driving the control sleeve **70**. As shown in FIG. **4**, the raceway surface **70***a* is formed inside the control sleeve **70** in a state in which a center point of the raceway surface **70***a* is eccentric with respect to a center of rotation of the control sleeve **70** as seen from the axial direction of the drive camshaft **44**. Accordingly, when the control sleeve **70** is rotated by the actuator **76** with the center of rotation of the control sleeve **70** as a center, the center point of the raceway surface **70***a* draws a circular locus as shown by the broken line in FIG. **4**.

Further, in the configuration example shown in FIG. **4**, a relative positional relation of the control sleeve **70** and the drive camshaft **44** is set so that the center of rotation of the drive camshaft **44** is located on the locus of the center point of the raceway surface **70***a* as seen from the axial direction of the drive camshaft **44**.

The above described control sleeves **70** are respectively disposed at each of the cylinders of the internal combustion engine **10** as shown in FIG. **2**. On an outer periphery of each of the control sleeves **70**, a gear **70***b* is formed over an entire circumference (only a part of the gear **70***b* is illustrated in FIG. **4** and FIG. **6** which will be described later).

Further, the variable intake valve operating apparatus **34** includes a control shaft **78** which is parallel with the drive camshaft **44** in the vicinity of the outer periphery of the control sleeve **70**. On a circumferential surface of the control shaft **78**, gears **78***a* which are meshed with the respective gears **70***b* of the respective, control sleeves **70** are provided. At one end part of the control shaft **78**, a gear **78***b* different from the gear **78***a* is formed.

The gear **78***b* of the control shaft **78** is meshed with a gear **80***b* formed at a distal end of an output shaft **80***a* of an electric motor (hereinafter, simply described as "motor") **80**. The control sleeve **70** is assumed to be rotatably supported by a cam housing (support member) not illustrated by using a predetermined fixing member (not illustrated). Further, the control shaft **78** is also assumed to be rotatably supported by the cam housing. In this connection, the motor **80** is driven on the basis of a command from an ECU **40** equipped with the internal combustion engine **10**.

As described above, the actuator **76** is configured by the gears **70***b* formed at the control sleeves **70**, the control shaft **78** which is connected to the control sleeves **70** via the gears **70***b* and the gears **78***a*, and the motor **80** which is connected to the control shaft **78** via the gear **78***b* and the gear **80***b*. According to the actuator **76** which is configured in this manner, a rotation position of the control sleeve **70** is adjusted by adjusting, on the basis of a command from the ECU **40**, a rotation position of the control shaft **78** by use of the motor **80**, and as a result, an eccentricity amount between the center of rotation of the drive camshaft **44** and a center of the raceway surface **70***a* can be adjusted.

(Definition of an Eccentric Angle φ)

Hereinafter, in the present specification, as the index for determining the eccentricity amount between the center of rotation of the drive camshaft **44** and the center of the raceway surface **70***a*, an "eccentric angle φ" will be used. Here, as shown in FIG. **4**, the eccentric angle φ is defined as an angle formed by a straight line which extends to the center of rotation of the drive camshaft **44** from the center of rotation of the control sleeve **70** and a straight line which extends to the center point of the raceway surface **70***a* from the center of rotation of the control sleeve **70**, as seen from the axial direction of the drive camshaft **44**. More specifically, in the state in which the center point of the raceway surface **70***a* and the center point of the drive camshaft **44** correspond to each other,

the eccentric angle ϕ is 0°. The eccentric angle ϕ is defined to be a larger value, as the center point of the raceway surface **70***a* rotates counterclockwise on the locus thereof to a larger extent in association with the rotation amount of the control sleeve **70** in the counterclockwise direction in FIG. **4** becoming larger. Further, in the state shown in FIG. **4** (more specifically, a state in which the center point of the raceway surface **70***a* is located in a position linearly symmetrical with the center of rotation of the drive camshaft **44** with the vertical line passing through the center of rotation of the control sleeve **70** set as a reference), the eccentric angle ϕ is 180°. The eccentricity amount of the center of rotation of the drive camshaft **44** and the center point of the raceway surface **70***a* becomes the maximum when the eccentric angle ϕ is 180°.

Further, below each of the driven cam lobes **50***a* of the respective cylinders, a rocker arm **82** is disposed for each intake valve **30** as shown in FIG. **2**. A rocker roller **82***a* which is in contact with the driven cam lobe **50***a* is rotatably mounted on a central part of the rocker arm **82**. Further, one end of the rocker arm **82** is supported by a valve stem of the intake valve **30**, and the other end of the rocker arm **82** is rotatably supported by a hydraulic lash adjuster **84**. The intake valve **30** is urged in a closing direction, that is, a direction to push up the rocker arm **82** by a valve spring **86**.

[Operation of the Variable Intake Valve Operating Apparatus]

FIG. **6** is a view expressing an operation of the link mechanism **68** for an interval in which the drive camshaft **44** makes one rotation (mainly a change of the rotation angle θ during one rotation of the drive camshaft **44**) as an example. More specifically, FIG. **6** is a view expressing the link mechanism **68** when the raceway surface **70***a* is in the same eccentric state as in the above described FIG. **4** (state in which the eccentric angle ϕ is 180°).

When the drive camshaft **44** rotates in the rotational direction of the drive camshaft **44** shown in FIG. **6**, the rotational force of the drive camshaft **44** is transmitted to the drive link **56** via the drive arm part **52***a* which is integrally fixed to the drive camshaft **44**. The rotational force of the drive camshaft **44** which is transmitted to the drive link **56** is transmitted to the driven cam lobe **50***a* which is formed integrally with the driven arm part **50***b* via the control roller side rotary shaft **62** and the driven link **60**. In this manner, the rotational force of the drive camshaft **44** is transmitted to the driven cam lobe **50***a* via the link mechanism **68**.

As a result, the respective elements of the link mechanism **68** and the driven cam lobe **50***a* are rotated in the same direction as the drive camshaft **44** synchronously with the rotation of the drive camshaft **44**. At this time, as already described, the control roller **64** rolls on the raceway surface **70***a* in the state in which the control roller **64** is always in contact with the raceway surface **70***a* in a contact point P, and revolves around the drive camshaft **44**.

The raceway surface **70***a* is a circumferential surface as already described. Therefore, when the center point of the raceway surface **70***a* corresponds to the center point of rotation of the drive camshaft **44** (when the eccentric angle ϕ is 0°), for an interval in which the control roller **64** makes one rotation on the raceway surface **70***a* in association with rotation of the drive camshaft **44**, the distance between the center of rotation of the drive camshaft **44** and the center of rotation of the control roller **64** does not change, and the relative rotation angle θ of the driven cam lobe **50***a* with respect to the drive camshaft **44** does not change, unlike the eccentric state as shown in FIG. **6**. Accordingly, in this case, the driven cam lobe **50***a* makes one rotation at an equal speed to the drive camshaft **44**.

Meanwhile, in a state in which the center point of the raceway surface **70***a* is eccentric from the center of rotation of the control sleeve **70** as shown in FIG. **6** as an example, a distance between the center of rotation of the control roller **64** which revolves around the drive camshaft **44** while being guided by the raceway surface **70***a* and the center of rotation of the drive camshaft **44** changes for an interval in which the drive camshaft **44** makes one rotation, and as a result, the rotation angle θ changes. Here, the contact point P of the control roller **64** to the raceway surface **70***a* in a timing in which a value θ0 at the time of the eccentric angle ϕ being 0° (see FIG. **7** which will be described later) and the rotation angle θ become equal to each other is referred to as an "equal rotation angle point P0" as shown in FIG. **6**(A) and the like.

(Definition of Narrow/Broad Zones)

The eccentric state shown in FIG. **6** is the state in which the rotation amount of the control sleeve **70** is regulated by the actuator **76** so that the eccentric angle ϕ becomes 180°, whereby the center point of the raceway surface **70***a* is located at a position linearly symmetrical with the center of rotation of the drive camshaft **44** with the vertical line passing through the center of rotation of the control sleeve **70** as a reference. In this state, when the control roller **64** is located at a substantially right half side of the raceway surface **70***a*, a distance between the center of rotation of the drive camshaft **44** and the center of rotation of the control roller **64** is made smaller than the distance at the time when the eccentric angle ϕ is 0° (more specifically, when the rotation angle θ is always 00). In the following description, a zone of the raceway surface **70***a* (zone at the substantially right half side in the case of FIG. **6**) in which the above described distance is made smaller than the distance at the time when the contact point P is at the equal rotation angle point P0 will be simply referred to as a "narrow zone".

When the above described distance is reduced as shown in FIG. **6**(A), the rotation angle θ is increased as compared with the value (θ0) at the time when the contact point P is at the equal rotation angle point P0. The rotational direction of the drive camshaft **44** is clockwise in FIG. **6**. Accordingly, when the above described rotation angle θ is increased in the above described narrow zone, the rotation position of the driven cam lobe **50***a* is advanced to the front side in the rotational direction of the drive camshaft **44**, as compared with that at the time when the contact point P is at the equal rotation angle point P0.

Meanwhile, when the control roller **64** is located at a substantially left half side of the raceway surface **70***a* in association with rotation of the drive camshaft **44** in the eccentric state shown in FIG. **6** (in the case of FIG. **6**(C)), the above described distance is increased as compared with that at the time when the contact point P is at the equal rotation angle point P0, contrary to the above description. In the following description, a zone of the raceway surface **70***a* in which the above described distance is increased (zone at the substantially left half side in the case of FIG. **6**) will be simply referred to as a "broad zone". In the broad zone, the above described rotation angle θ decreases as compared with that at the time when the contact point P is at the equal rotation angle point P0. As a result, in the above described broad zone, the rotation position of the driven cam lobe **50***a* is delayed to the rear side in the rotational direction of the drive camshaft **44** as compared with that at the time when the contact point P is at the equal rotation angle point P0.

An operation of the link mechanism **68** during one rotation of the drive camshaft **44** shown in each view in FIG. **6** will be described in more detail. In a timing in which the contact point P between the control roller **64** and the raceway surface

70*a* is located in the narrow zone as shown in FIG. **6**(A), the relative rotation angle θ of the driven cam lobe **50***a* to the drive camshaft **44** increases with respect to the value θ0 in the case of the eccentric angle ϕ being 0° (equal speed time). Further, in a timing shown in FIG. **6**(B), the contact point P is located in a position near the aforementioned equal rotation angle timing P0, and therefore, the rotation angle θ is close to the value in the case of the eccentric angle ϕ being 0° (equal speed time). Further, in a timing in which the contract point P is located in the broad zone as shown in FIG. **6**(C), the above described rotation angle θ reduces with respect to the value θ0 at the equal speed time. Thereafter, in a timing in which the contact point P enters the narrow zone again from the broad zone as shown in FIG. **6**(D), the rotation angle θ increases again. As above, according to the variable intake valve operating apparatus **34** of the present embodiment, the center of rotation of the drive camshaft **44** and the center of the raceway surface **70***a* are made eccentric from each other by rotating the control sleeve **70** (more specifically, the eccentric angle ϕ is set to the angle other than 0°), whereby the relative rotation angle θ of the driven cam lobe **50***a* to the drive camshaft **44** for an interval in which the drive camshaft **44** makes one rotation can be made variable. The rotation angle θ can be changed in this manner, whereby the rotational speed of the driven cam lobe **50***a* for an interval in which the drive camshaft **44** makes one rotation can be increased and decreased with respect to the rotational speed of the drive camshaft **44**.

(Definition Acceleration/Deceleration Zone)

In the eccentric state shown in FIG. **6**, when the contact point P of the control roller **64** is located at the point P1 in the narrow zone at the substantially right half side of the raceway surface **70***a*, the distance between the center of rotation of the drive camshaft **44** and the center of rotation of the control roller **64** becomes the smallest, and the rotation angle θ between the drive shaft and the driven shaft is the most increased. Meanwhile, in this eccentric state, when the contact point P of the control roller **64** is located at a point P2 in the broad zone at the substantially left half side of the raceway surface **70***a*, the distance between the center of rotation of the drive camshaft **44** and the center of rotation of the control roller **64** becomes the largest, and the rotation angle θ between the drive shaft and the driven shaft is the most reduced. More specifically, while the contact point P of the control roller **64** moves toward the point P1 from the point P2, the change amount of the rotation angle θ per unit cam angle increases, and therefore, the rotational speed of the driven cam lobe **50***a* becomes higher than the rotational speed of the drive camshaft **44** (accelerates), whereas while the contact point P of the control roller **64** moves toward the point P2 from the point P1, the change amount of the rotation angle θ per unit cam angle decreases, and therefore, the rotational speed of the driven cam lobe **50***a* becomes lower than the rotational speed of the drive camshaft **44** (decelerates). Therefore, in the following description, the zone from the point P2 to the point P1 on the raceway surface **70***a* will be simply described as an "acceleration zone", and the zone from the point P1 to the point P2 on the raceway surface **70***a* will be simply described as a "deceleration zone".

FIG. **7** is a schematic view for explaining the operations of the variable intake valve operating apparatus **34** when the eccentric angle ϕ is changed by 90° at a time. FIG. **8** is a diagram expressing a tendency of a change of valve opening characteristics of the intake valve **30** in association with a change of the eccentric angle ϕ in the variable intake valve operating apparatus **34**. More specifically, FIG. **8**(A) is a diagram expressing a tendency of the change of the operation angle of the intake valve **30** in association with the change of the eccentric angle ϕ, and FIG. **8**(B) is a diagram expressing a tendency of a change of an opening timing of the intake valve **30** in association with the change of the eccentric angle ϕ. Further, FIG. **9** is a diagram expressing each lift curve of the intake valve **30** which is obtained when the eccentric angle ϕ is changed by 90° at a time. Each drawing in FIG. **7** expresses an operation state of the variable intake valve operating apparatus **34** (particularly, the link mechanism **68**) in the timing at which the contact point with the rocker roller **82***a* in the driven cam lobe **50***a* is switched to the nose part **50***a***2** from the base circle part **50***a***1** (more specifically, timing at which the opening timing of the intake valve **30** arrives).

First, an operation state shown in FIG. **7**(A) is the state in which the eccentric angle ϕ is 0°, that is, the state in which the center of rotation of the drive camshaft **44** and the center of the raceway surface **70***a* correspond to each other. In this case, the driven cam lobe **50***a* makes one rotation at an equal speed to the drive camshaft **44** as described above. Further, the operation angle of the intake valve **30** which is obtained in this case will be set as "OA1" hereinafter as shown in FIG. **8** for convenience of explanation.

Next, an operation state shown in FIG. **7**(B) (eccentric angle ϕ=90° will be described. The operation state is obtained by rotating the control sleeve **70** by 90° in a counterclockwise direction in FIG. **7** with respect to the operation state shown in FIG. **7**(A). When the "acceleration zone" and the "deceleration zone" which are defined as described above are applied to the operation state shown in FIG. **7**(B), the result is as shown in FIG. **7**(B). Here, when the operation angle of the intake valve **30** in the reference state (state at the time of equal speed shown in FIG. **7**(A)) is an ordinary value (value larger than 180° by a predetermined amount in crank angle), the cam angle becomes a value with a value of half the above described predetermined amount being added to 90° in the time period in which the nose part **50***a***2** of the driven cam lobe **50***a* presses the rocker roller **82***a*. As a result, the control roller **64** in the lift zone of the intake valve **30** substantially passes through the acceleration zone. Therefore, the operation angle of the intake valve **30** in the operation state shown in FIG. **7**(B) becomes smaller than that in the equal speed state shown in FIG. **7**(A), as shown in FIG. **8**(A) and FIG. **9**.

More specifically, as the eccentric angle ϕ is changed toward 90° from 0°, the operation angle of the intake valve **30** becomes gradually smaller as shown in FIG. **8**(A), with increase of the eccentricity amount between the center of rotation of the drive camshaft **44** and the center of rotation of the raceway surface **70***a* under the situation in which the lift zone of the intake valve **30** and the acceleration zone of the control roller **64** overlay each other. However, the tendency of the change of the operation angle of the intake valve **30** in association with change of the eccentric angle ϕ changes in accordance with setting of the respective components of the variable intake valve operating apparatus **34** (the angle formed by the driven cam lobe **50***a* and the driven arm part **50***b* (driven shaft), the ratio of the lengths of the respective links of the link mechanism **68**, or the like). In the case of setting of the variable intake valve operating apparatus **34** shown in FIG. **7**, when the eccentric angle θ is a value around 90°, a minimum operation angle OAmin within a variable range of the operation angle in the variable intake valve operating apparatus **34** is obtained as the opening angle of the intake valve **30**, as shown in FIG. **8**(A).

Further, in the operation state shown in FIG. **7**(B), the contact point P of the control roller **64** is located in the broad zone (zone in which the rotation angle θ becomes smaller than the rotation angle θ0 at the time of the equal rotation angle point P0) in the timing at which the opening timing of

the intake valve **30** arrives. Therefore, the opening timing of the intake valve **30** in this operation state becomes a value at the retard side with respect to the value at the time of equal speed as shown in FIG. **8**(B) and FIG. **9**. Further, in the case of the setting of the present variable intake valve operating apparatus **34**, when the eccentric angle ϕ is changed toward 90° from 0°, the retard amount of the opening timing of the intake valve **30** decreases after the retard amount temporarily increases as shown in FIG. **8**(B).

Next, an operation state shown in FIG. **7**(C) (eccentric angle ϕ=180°) will be described. The operation state is obtained by further rotating the control sleeve **70** by 90° in the counterclockwise direction in FIG. **7**, with respect to the operation state shown in FIG. **7**(B). When the "acceleration zone" and the "deceleration zone" are applied to the operation state shown in FIG. **7**(C), the result is as shown in FIG. **7**(C). More specifically, the contact point P of the control roller **64** to the raceway surface **70***a* is located in the acceleration zone at the initial time of start of lift operation of the intake valve **30**, and is thereafter switched to the deceleration zone from halfway through the lift operation. Especially in the case of the setting of the variable intake valve operating apparatus **34** shown in FIG. **7**, the action of acceleration in the first half portion of the lift zone of the intake valve **30** and the action of deceleration in the latter half portion thereof cancel out each other, and as a result, the operation angle of the intake valve **30** becomes the same value OA1 as that at the time of equal speed as shown in FIG. **8**(A).

Further, as the eccentric angle ϕ is changed toward 180° from 90°, the ratio of the deceleration zone in the lift zone of the intake valve **30** increases, and therefore, the operation angle of the intake valve **30** becomes larger as shown in FIG. **8**(A).

Further, in the operation state shown in FIG. **7**(C), at a timing at which the opening timing of the intake valve **30** arrives, the contact point P of the control roller **64** is located in the narrow zone (zone in which the rotation angle θ becomes larger than the rotation angle θ0 at the time of the equal rotation angle point P0). Therefore, the opening timing of the intake valve **30** in this operation state has a value at an advance side with respect to the value at the time of equal speed, as shown in FIG. **8**(B) and FIG. **9**. Further, in the case of the setting of the present variable intake valve operating apparatus **34**, when the eccentric angle ϕ is changed toward 180° from 90°, the advance amount of the opening timing of the intake valve **30** gradually increases as shown in FIG. **8**(B).

Next, an operation state (eccentric angle ϕ=270° shown in FIG. **7**(D) will be described. This operation state is obtained by further rotating the control sleeve **70** by 90° in the counterclockwise direction in FIG. **7** with respect to the operation state shown in FIG. **7**(C). When the "acceleration zone" and the "deceleration zone" are applied to the operation state shown in FIG. **7**(D), the result is as shown in FIG. **7**(D). As a result, the control roller **64** in the lift zone of the intake valve **30** is located in the acceleration zone at the initial time of start of lift, but mainly passes through the deceleration zone. Therefore, the operation angle of the intake valve **30** in the operation state shown in FIG. **7**(D) becomes larger than that in the equal speed state shown in FIG. **7**(A).

More specifically, when the eccentric angle ϕ is changed toward 270° from 180°, the ratio of the deceleration zone in the lift zone of the intake valve **30** increases, and therefore, the operation angle of the intake valve **30** becomes gradually large as shown in FIG. **8**(A). In the case of the setting of the variable intake valve operating apparatus **34** shown in FIG. **7**, when the eccentric angle ϕ is at a value around 270°, a maximum operation angle OAmax within the variable range of the operation angle in the variable intake valve operating apparatus **34** is obtained as the opening angle of the intake valve **30**, as shown in FIG. **8**(A).

Further, in the operation state shown in FIG. **7**(D), at the timing at which the opening timing of the intake valve **30** arrives, the contact point P of the control roller **64** is located in the narrow zone (the zone in which the rotation angle θ becomes larger than the rotation angle θ0 at the time of the equal rotation angle point P0). Therefore, the opening timing of the intake valve **30** in this operation state is at a value at the advance side with respect to the value at the time of equal speed, substantially similarly to the case of 180°, as shown in FIG. **8**(B) and FIG. **9**. Further, in the case of the setting of the present variable intake valve operating apparatus **34**, when the eccentric angle ϕ is changed toward 270° from 180°, the advance amount of the opening timing of the intake valve **30** temporarily increases and thereafter, decreases, as shown in FIG. **8**(B).

Finally, the operation of the variable intake valve operating apparatus **34** when the eccentric angle ϕ changes to 360° (0°) from 270°, that is, when the operation state is returned to the operation state shown in FIG. **7**(A) from the operation state shown in FIG. **7**(D) will be described. As the eccentric angle ϕ is changed toward 360° from 270°, the operation angle of the intake valve **30** becomes gradually smaller toward the value OA1 at the time of equal speed as shown in FIG. **8**(A) with decrease in the eccentricity amount between the center of rotation of the drive camshaft **44** and the center of the raceway surface **70***a* under the situation in which the lift zone of the intake valve **30** and the deceleration zone of the control roller **64** overlay each other. Further, as for the advance amount of the opening timing of the intake valve **30** when the eccentric angle ϕ changes to 360° (0°) from 270°, the difference from the value at the time of equal speed also becomes gradually smaller with decrease of the eccentricity amount.

As described above, according to the variable intake valve operating apparatus **34** of the present embodiment, the eccentric angle ϕ is changed by rotationally driving the control sleeve **70** including the raceway surface **70***a* the center of which is eccentric with respect to the center of rotation of the control sleeve **70**, whereby the distance between the center of rotation of the drive camshaft **44** and the center of rotation of the control roller **64** is changed. As a result, the relative rotation angle θ of the driven cam lobe **50***a* with respect to the drive camshaft **44** for an interval in which the drive camshaft **44** makes one rotation changes. In other words, the center of revolution of the control roller **64** which revolves around the drive camshaft **44** by rolling along the raceway surface **70***a* is changed in accordance with the eccentric angle ϕ which is adjusted like this. Consequently, according to the present variable intake valve operating apparatus **34**, the rotational speed of the driven cam lobe **50***a* for an interval in which the drive camshaft **44** makes one rotation can be continuously increased and decreased with respect to the rotation angle of the drive camshaft **44**, in accordance with the control position of the raceway surface **70***a* (rotation position of the control sleeve **70**) in association with adjustment of the eccentric angle ϕ by the actuator **76**. Thereby, the operation angle of the intake valve **30** can be made continuously variable in accordance with the control position of the raceway surface **70***a* as shown in FIG. **8**(A) and FIG. **9**. According to the variable intake valve operating apparatus **34**, if the VVT mechanism **48** is also used in addition to the control of the eccentric angle ϕ, the opening timing of the intake valve **30** can be regulated to be in any timing with respect to the value obtained by control of the eccentric angle ϕ.

According to the variable intake valve operating apparatus **34** which is configured as above, the operation angle of the intake valve **30** can be made continuously variable in the following characteristic mode. More specifically, the operation angle can be made variable in both directions of a reducing direction and an increasing direction, in the mode in which when the control sleeve **70** is rotated in one direction, the operation angle of the intake valve **30** becomes large after becoming small, and thereafter, becomes small again. On top of it, according to the present variable intake valve operating apparatus **34**, in what timing in the lift zone, the acceleration/deceleration zones arrive gradually becomes different, and the acceleration and deceleration amounts gradually becomes different, when the control sleeve **70** is rotated in one direction and the eccentric angle ϕ is continuously changed. While the relation between the lift zone and the acceleration/deceleration zones changes as above, and the acceleration and deceleration amounts in the lift zone change, the eccentric angles ϕ at which the operations of acceleration and deceleration as the total of adding up the operation of acceleration and the operation of the deceleration in the lift zone are equal (that is, the operation angles are equal) appear as two points like, for example, 0° and 180°. As a result, the operation angle can be said to increase and decrease with respect to the change in one direction of the eccentric angle ϕ in the curve as shown in FIG. **8**(A). However, the control sleeve **70** for making the operation angle variable is not limited to the one that is driven in one predetermined direction, but may be driven in both directions (reciprocating direction) in accordance with necessity.

Further, in the variable intake valve operating apparatus **34** of the present embodiment, the raceway surface **70***a* is formed into a circumferential surface, and the relative positional relation between the control sleeve **70** and the drive camshaft **44** is set so that the center of rotation of the drive camshaft **44** is located on the locus of the raceway surface **70***a* at the time of rotation of the control sleeve **70**. Thereby, the control sleeve **70** is rotated so that the center of the raceway surface **70***a* corresponds to the center of rotation of the drive camshaft **44**, whereby the operation state in which the driven cam lobe **50***a* for an interval in which the drive camshaft **44** makes one rotation has equal speed can be obtained.

Further, as the effect which can be provided based on the configuration of the variable intake valve operating apparatus **34**, the following effect is cited.

More specifically, a spring reaction force from the valve spring **86** acts on, via the driven cam lobe **50***a*, the link mechanism **68** which revolves around the drive camshaft **44**. The spring reaction force acts in the radial direction of the control sleeve **70** via the control roller **64** and the raceway surface **70***a*. Further, an inertia force from the link mechanism **68** in association with rotation of the drive camshaft **44** acts on the raceway surface **70***a* in the radial direction of the control sleeve **70** via the control roller **64**. In the variable intake valve operating apparatus **34** of the present embodiment described above, the direction in which the control sleeve **70** is operated to change the operation angle of the intake valve **30** differs from the action direction (radial direction) of the above described spring reaction force and the like, and is the rotational direction (circumferential direction) of the control sleeve **70**. Accordingly, even if the control sleeve **70** is rotationally driven in the direction against the component force in the circumferential direction of the above described spring reaction force and the like, the load torque of the actuator **76** in rotationally driving the control sleeve **70** to make the operation angle variable becomes small torque based on the component force in the circumferential direction. Therefore, according to the present variable intake valve operating apparatus **34**, the load torque which acts on the actuator **76** can be significantly reduced.

Further, according to the variable intake valve operating apparatus **34** of the present embodiment, as the contact member which is in contact with the raceway surface **70***a*, the control roller **64** is adopted, and is rolled on the raceway surface **70***a*. Therefore, as compared with the case in which a member which performs contact with the raceway surface **70***a* by using slide is adopted as the above described contact member, friction and wear of the contact parts can be reduced.

Further, in the variable intake valve operating apparatus **34** of the present embodiment, the driven link **60** is not disposed at the rear side, but is disposed at the front side in the rotational direction of the drive camshaft **44** with respect to the drive arm part **52***a* and the drive link **56** with the control roller **64** interposed therebetween. According to the configuration like this, the force which is caused by the above described spring reaction force of the valve spring **86** and acts on the drive link **56** and the driven link **60** becomes a compression force instead of a tensile force and a bending force. Therefore, deformation and stress of the drive link **56** and the driven link **60** can be reduced, and the position of the control roller **64** (the rotation angle θ of the drive camshaft **44** and the driven cam lobe **50***a*) is more reliably determined.

(Relation Between the Lift Zone of the Intake Valve **30** and Acceleration/Deceleration Zones)

Next, with reference to FIG. **10**, previously setting elements which give a main influence on the valve opening characteristics (shape of the lift curve) of the intake valve **30** which are realized by the variable intake valve operating apparatus **34** will be described.

FIG. **10** is a diagram expressing a relation between the lift zone of the intake valve **30** and the acceleration/deceleration zones. More specifically, the relation shown in FIG. **10** is the one in the eccentric state in which the center point of the raceway surface **70***a* is away from the center of rotation of the drive camshaft **44** (state in which the raceway surface **70***a* is controlled to be at a certain eccentric angle ϕ other than 0°). Further, the vertical axis of FIG. **10** represents a change amount of the rotation angle θ for an interval in which the drive camshaft **44** makes one rotation (corresponding to a change of a rotational speed (angular speed) of the driven cam lobe **50***a* with the rotational speed of the drive camshaft **44** (equal speed) as the reference).

When the moving range of the control roller **64** on the raceway surface **70***a* in the lift zone of the intake valve **30** changes in the eccentric state of the raceway surface **70***a* with the eccentric angle ϕ being other than 0° as shown in FIG. **10**, setting of the relation between the lift zone and the acceleration/deceleration zones changes as the three cases illustrated as shown FIG. **10**. This brings about changes in how the rotational speed of the driven cam lobe **50***a* is increased or decreased in the lift zone, and how the acceleration or deceleration amount of the driven cam lobe **50***a* in the lift zone changes. Therefore, the lift curve of the intake valve **30** changes in accordance with setting of the lift zone as shown in FIG. **10**. For example, in the case of (i) in FIG. **10**, the entire range of the lift zone belongs to the deceleration zone, and therefore, the lift curve is obtained, in which the change of the lift amount entirely becomes slow with respect to the lift curve at the time of the equal speed. In the case of (ii) in FIG. **10**, the zone is switched from the deceleration zone to the acceleration zone halfway through the lift zone, and therefore, the lift curve is obtained, in which the change of the lift amount at the opening side becomes slow and the change of the lift amount at the closing side becomes fast with respect to

the lift curvet at the time of the equal speed. Further, in the case of (iii) in FIG. **10**, the entire range of the lift zone belongs to the acceleration zone, and therefore the lift curve is obtained, in which the change of the lift amount entirely becomes fast with respect to the lift curve at the time of the equal speed.

Here, the position (moving range) of the control roller **64** on the raceway surface **70***a* in the relation with the lift zone of the intake valve **30** can be changed by adjusting, for example, the mounting angle between the driven cam lobe **50***a* and the driven arm part **50***b* (driven shaft) on the cam piece **50**. As a result, as illustrated as the three cases in FIG. **10**, setting of the relation between the lift zone and the acceleration/deceleration zones can be changed.

On top of it, the shape of the pantographic link mechanism **68** determines the acceleration and deceleration amounts of the driven cam lobe **50***a* for an interval in which the drive camshaft **44** makes one rotation. The lift curve of the intake valve **30** is determined depending on in which part of the lift zone acceleration and deceleration are performed. More specifically, the factors which determine the acceleration and deceleration amounts of the driven cam lobe **50***a* are the length of the side of each link of the link mechanism **68**, the eccentricity amount between the center of rotation of the drive camshaft **44** and the center of the raceway surface **70***a* (the moving amount of the raceway surface **70***a* in the plane direction orthogonal to the axial line of the drive camshaft **44**) and the like. Further, the factors which determine setting of the relation between the lift zone and the acceleration/deceleration zones are the ratio of the length of the side of each link of the link mechanism **68**, the shape of the locus (FIG. **4** and the like) of the center point of the raceway surface **70***a* and the like, in addition to the above described mounting angle. Further, the acceleration/deceleration zones of the driven cam lobe **50***a* for an interval in which the drive camshaft **44** makes one rotation become opposite depending on whether the driven cam lobe **50***a* is located at the front side in the rotational direction of the drive camshaft **44** with respect to the drive camshaft **44** as the configuration of the present embodiment shown in FIG. **4** and the like, or the driven cam lobe **50***a* is located at the rear side thereof conversely.

[Control of the Internal Combustion Engine Using the Variable Valve Operating Apparatus in Embodiment 1]

Next, with reference to FIG. **11** to FIG. **16**, characteristic control of the internal combustion engine **10** using the variable intake valve operating apparatus **34** which has the configuration described above will be described.

FIG. **11** is a diagram expressing a relation between the eccentric angle ϕ and each lift curve of the intake valve **30** which is to be changed in accordance with the operation condition of the internal combustion engine **10** in embodiment 1 of the present invention.

According to the variable intake valve operating apparatus **34** having the configuration described above, the control sleeve **70** is rotationally driven and the eccentric angle ϕ is changed, whereby the lift speed in each timing in the lift zone changes. As a result, the operation angle of the intake valve **30** can be changed while a timing at which the lift amount shows a peak (hereinafter, sometimes simply abbreviated as a "lift peak timing") is changed. Furthermore, in the present variable intake valve operating apparatus **34**, the raceway surface **70***a* the center of which is eccentric with respect to the center of rotation of the control sleeve **70** which is rotationally driven by the actuator **76** is included, and rolling of the control roller **64** of the pantographic link mechanism **68** is guided by the raceway surface **70***a*. Thereby, as already described, the operation angle of the intake valve **30** can be made continuously variable in the characteristic mode in which the operation angle of the intake valve **30** is made variable in both directions of the reducing direction and the increasing direction as also shown in FIG. **11** when the control sleeve **70** is rotated in one direction.

Thus, in the present embodiment, the eccentric angle ϕ is changed in accordance with the operation conditions of the internal combustion engine **10** by using the aforementioned characteristic variable intake valve operating apparatus **34**, whereby the lift curves of the intake valve **30** with the lift peak timings differing in accordance with the operation conditions are discriminatingly used as shown in FIG. **11**.

More specifically, in the present embodiment, at the time of a first high engine speed which is higher than a first predetermined engine speed NE1 and is not higher than a second predetermined engine speed NE2 (at the time of use of a high rotational speed area which is assumed to be used during a normal operation of the internal combustion engine **10**), 0° is selected as the eccentric angle ϕ which is a control amount of the actuator **76**. The lift curve in the case of the eccentric angle ϕ being 0° is the one that is obtained when the driven cam lobe **50***a* rotates at an equal speed to the drive camshaft **44**. In the present embodiment, the profile of the driven cam lobe **50***a* is set so that the lift peak timing of the lift curve which is obtained in this case is at a closing side with respect to the center of the valve opening time period (lift zone) (so that the intake air flow rate can be enhanced efficiently at the time of the first high engine speed).

Further, in the present embodiment, at the time of a low engine speed (except for an acceleration request time) which is not higher than the above described first predetermined speed NE1, 180° is selected as the eccentric angle ϕ. When the eccentric angle ϕ is 180°, action of acceleration is exerted in the first half part of the lift zone, and action of deceleration is exerted in the following latter half part, as already described. As a result, the lift peak timing of the lift curve which is obtained in this case moves to the opening side as compared with the case of the eccentric angle ϕ being 0°, and is in a substantially center of the valve opening time period (timing which is slightly close to the closing side) as shown in FIG. **11**. Further, in this case, as the entire valve opening time period, the action of acceleration and the action of deceleration cancel out each other, and the operation angle of the intake valve **30** becomes the same value OA1 as in the case of the eccentric angle ϕ being 0° (the equal speed time).

Further, in the present embodiment, when an acceleration request (high load request) is issued in the above described low engine speed time, the eccentric angle ϕ is changed to 210° (one example) from 180°. In the lift curve in the case of the eccentric angle ϕ being set as 210°, the rotational speed of the driven cam lobe **50***a* is relatively decreased in the initial time of start of lift of the intake valve **30**, as compared with the case of the eccentric angle ϕ being 180°. As a result, the lift amount in the predetermined zone in the vicinity of the opening timing of the intake valve **30** is small as compared with the case of the eccentric angle ϕ being 180°. Furthermore, as compared with the eccentric angle ϕ being 180°, the lift peak timing moves to the closing side, and the operation angle becomes large (OA2).

Furthermore, in the present embodiment, at a time of a second high engine speed which is higher than the above described second predetermined engine speed NE2 (at the time when the engine speed irregularly increases or the like), 320° (one example) is selected as the eccentric angle ϕ. When the eccentric angle is 320°, the rotational speed of the driven cam lobe **50***a* is decreased in the substantially entire lift zone including a predetermined zone (a ramp section at the closing

side) in the vicinity of the closing timing of the intake valve **30** as compared with the case of the eccentric angle φ being 0°. As a result, as compared with the case of the eccentric angle φ being 0°, the lift peak timing moves to the closing side, and the operation angle becomes large (OA3).

FIG. **12** is a flow chart showing a control routine which the ECU **40** executes in order to realize the control of the variable intake valve operating apparatus **34** described above in accordance with the operation conditions of the internal combustion engine **10**. The present routine is repeatedly executed at each predetermined control period.

In the routine shown in FIG. **12**, it is determined whether or not the engine is in the time of the first high engine speed in which the present engine speed is higher than the first predetermined engine speed NE1 and is not higher than the second predetermined engine speed NE2 (NE2>the first predetermined engine speed NE1) based on the output of the crank angle sensor **38** (step **100**). When it is determined that the engine is in the time of the first high engine speed as a result, 0° is used as the eccentric angle φ (step **102**).

Meanwhile, when it is determined that the engine is not in the time of the first high engine speed in the above described step **100**, it is determined whether or not the engine is in the low engine speed time in which the present engine speed is not higher than the above described first predetermined engine speed NE1 (step **104**). When it is determined that the engine is in the low engine speed time as a result, next, it is determined whether or not an acceleration request (high load request) is issued from a driver based on the accelerator pedal position which is detected by the accelerator position sensor **42** (step **106**).

When the determination of the above described step **106** is not established, that is to say, at the time of the low engine speed without the acceleration request being issued, 180° is used as the eccentric angle φ (step **108**). Meanwhile, when the determination of the above described step **106** is established, that is to say, when it is determined that an acceleration request is issued at the time of the low engine speed, 210° is used as the eccentric angle φ (step **110**).

Meanwhile, when the determination of the above described step **104** is not established, next, it is determined whether or not the present engine speed is higher than the above described second predetermined engine speed NE2 (step **112**). When the determination of the present step **112** is established as a result, 320° is used as the eccentric angle φ (step **114**). The second predetermined engine speed NE2 is the value which is set in advance as the threshold value which enables determination of whether or not the present engine speed is the engine speed which can ensure normal mobility of the variable intake valve operating apparatus **34** at the time of a high engine speed. Further, the processing of the present step **114** may be also applied to the variable exhaust valve operating apparatus **36**.

According to the control routine shown in FIG. **12** described above, the eccentric angle φ is changed in accordance with the operation conditions of the internal combustion engine **10**, whereby the lift curves of the intake valve **30** with different lift peak timings can be discriminatingly used in accordance with the operation conditions. Next, the effect which is obtained by each control of the eccentric angle φ in accordance with the operation conditions of the internal combustion engine **10** will be described.

First, with reference to FIG. **13** and FIG. **14**, control of switching the eccentric angle φ between 180° and 0° in accordance with the value of the engine speed will be described.

FIG. **13** is a view comparing the lift curves of the intake valve **30** and the flow rates of the intake air which is taken into the cylinder during the lift zone in the case of the eccentric angle φ being 0° and in the case of the eccentric angle φ being 180°. In FIG. **13**(A), for comparison, the opening timings of the two lift curves are matched with each other by using the VVT mechanism **48**. Further, FIG. **13**(B) expresses a difference of the intake air flow rates due to a difference of the above described two lift curves in the same operation condition (at the time of the first high engine speed).

Here, the conventional variable valve operating apparatus in which the operation angle of the valve is made variable on the precondition that the drive camshaft is rotationally driven by using the rotational force of the crankshaft differs from the variable intake valve operating apparatus **34** of the present embodiment, and is the one in which when the actuator (control shaft) is controlled in one direction, the operation angle of the valve increases unidirectionally with the lift amount, and when the actuator is controlled in the opposite direction, the operation angle of the valve decreases unidirectionally with the lift amount as described in, for example, Japanese Laid-Open Patent Application Publication No. 2009-57868. Accordingly, in the conventional variable valve operating apparatus like this, the lift curve with only one characteristic (one shape) can be obtained for one operation angle. This is not limited to the variable valve operating apparatus which makes the operation angle of the valve continuously variable as the one described in the above described publication, but also applies to the variable valve operating apparatus which makes the operation angle of the valve variable stepwise.

In contrast with this, according to the variable intake valve operating apparatus **34** of the present embodiment, the lift curve φ**32** 180°) which is used at the time of the low engine speed and the lift curve (φ=0°) which is used at the time of the first high engine speed are equal in the operation angle and differ in the lift peak timing as already described and shown in FIG. **13**(A). Further, in the present embodiment, as shown in FIG. **13**(A) by being enlarged, the lift curve (φ=0°) which is used at the time of the first high engine speed is set to have a lower lift amount of the predetermined section in the vicinity of the closing timing than the lift curve (φ=180°) which is used at the time of the low engine speed.

FIG. **14** is a view for explaining a behavior of intake air in an intake stroke and a compression stroke.

As the measures for enhancing the output power of the internal combustion engine **10**, it is effective to enhance torque by increase of the intake air amount. When air flows into the cylinder as shown in FIG. **14**(A) during the lift zone of the intake valve **30**, timing in which much air easily enter is present due to the influence of the pulsation of the intake air and the position of the piston **12**. Such timing changes in accordance with the engine speed. More specifically, as the engine speed becomes higher, the above described timing becomes relatively later. Further, when the engine speed becomes high, the inertia force of the air taken into the cylinder becomes large, and therefore, even after the piston passes the intake bottom dead center (BDC), air flows into the cylinder as shown in FIG. **13**(B). However, if the intake valve **30** is kept in the opened state for a long time after entering the compression stroke, the amount of air which is charged in the cylinder decreases due to blowback to the intake passage **16** side. Accordingly, the closing timing of the intake valve **30** needs to be determined in consideration of the influence of the inertia force of air and prevention of blowback, as shown in FIG. **14**(B).

However, in the case of the conventional variable valve operating apparatus of the configuration in which the operation angle of the intake valve changes in only one direction (any one of the increasing direction and the decreasing direc-

tion) with the lift amount when the actuator is controlled in one direction, as in the variable valve operating apparatus of the configuration shown in the above described publication, the following problem occurs. More specifically, if the operation angle and the lift amount of the intake valve are increased with increase of the engine speed in order to obtain a high lift amount in timings at which much air easily enters, the closing timing of the intake valve is delayed, and therefore, the charge air amount in the cylinder cannot be favorably ensured due to the influence of blowback as a result. Conversely, if the closing timing of the intake valve is set to a timing in which blowback does not occur, a sufficient lift amount cannot be ensured in timings in which much air easily enters.

In contrast with this, according to the variable intake valve operating apparatus **34** of the present embodiment, the lift peak timing of the lift curve (ϕ=0°) at the time of the first high engine speed which is used at a higher engine speed side than the lift curve at the time of the low engine speed (ϕ=180°) can be deviated to the closing side in agreement with increase in the engine speed. Thereby, a large lift amount can be ensured in the timings in which much air easily enters irrespective of the value of the engine speed, as shown in FIG. **13**(A) and FIG. **13**(B) by being enlarged, and therefore, the charge air amount in the cylinder can be enhanced. Further, even when the lift peak timings are deviated between the two lift curves as described above, decrease of the charge air amount in the cylinder due to the influence of blowback of the intake air as in the case of the conventional variable valve operating apparatus can be prevented.

Further, in the lift curve at the time of the first high engine speed (ϕ=0°), the lift amount of the predetermined section in the vicinity of the closing timing is set to be small as compared with the lift curve at the time of the low engine speed (ϕ=180°) as shown in FIG. **13**(A) by being enlarged, whereby blowback of the intake air also can be decreased more as shown in FIG. **13**(B) by being enlarged.

As above, the eccentric angle ϕ is changed between 180° and 0° in accordance with the value of the engine speed, whereby the output performance can be favorably enhanced by torque enhancement of the internal combustion engine **10** in association with increase in the intake air flow rate.

Next, control of selecting 210° as the eccentric angle when an acceleration request is issued at the time of the low engine speed will be described with reference to FIG. **15**.

FIG. **15** is a diagram which compares the two lift curves with the eccentric angles ϕ of 180° and 210°, in relation with a valve stamp area.

In order to increase a valve overlap amount with the valve opening time period of the exhaust valve **32** and the valve opening time period of the intake valve **30** overlapping each other, it is effective to advance the opening timing of the intake valve **30** with respect to the intake and exhaust top dead center. However, if the advance amount of the opening timing of the intake valve **30** is too large, interference of the piston **12** and the intake valve **30** (so-called valve stamp) is feared. The area shown by applying hatching in FIG. **15** shows "valve stamp area" of the lift amount of the intake valve **30** in which valve stamp occurs.

In the lift curve at the time of the eccentric angle ϕ being 210°, the lift amount in the predetermined section in the vicinity of the opening timing of the intake valve **30** is small as already descried, with respect to the lift curve at the time of the eccentric angle ϕ being 180° (the lift curve in which the lift peak timing is in the substantially center of the valve opening time period). Therefore, as compared with the lift curve at the time of the eccentric angle ϕ being 180°, a long valve opening time period until reaching the valve stamp area from the opening timing can be secured as shown in FIG. **15**. Thereby, when 210° is selected as the eccentric angle ϕ, the opening timing of the intake valve **30** (more specifically, the entire lift curve) can be advanced more significantly by using the VVT mechanism **48**, as compared with the case in which 180° is selected.

In the present embodiment, when an acceleration request (high load request) is issued in the above described low engine speed time, 210° is selected as the eccentric angle ϕ. Thereby, the valve overlap amount is effectively increased at the time of acceleration. Therefore, in the internal combustion engine **10** including the turbo supercharger **22**, a scavenging effect is enhanced by increase of the valve overlap amount in the state in which the intake pressure becomes higher than the exhaust pressure at the time of acceleration. As a result, the supercharging pressure can be effectively enhanced, and therefore the output performance of the internal combustion engine **10** can be enhanced. Further, if it is under the situation in which the intake pressure is lower than the exhaust pressure at the time of acceleration, an internal EGR gas amount is increased by increase of the valve overlap amount. As a result, the fuel consumption performance and exhaust emission performance of the internal combustion engine **10** can be enhanced. In the lift curve in the example shown in FIG. **15** (ϕ=210°), the rotational speed of the driven cam lobe **50***a* increases at the closing side, and the rate of acceleration at the time of seating of the intake valve **30** becomes high. However, the time of using the present lift curve is the time of an acceleration request from the time of the low engine speed in which the rotational speed of the drive camshaft **44** is low, and therefore, it can be said that no problem arises in securing normal mobility of the variable intake valve operating apparatus **34**.

Next, with reference to FIG. **16**, control of selecting 320° as the eccentric angle ϕ at the time of the second high engine speed will be described.

FIG. **16** is a diagram which compares the lift curve with the eccentric angle ϕ of 320° with the lift curve (ϕ=0° at the time of the equal speed.

As already described and as shown in FIG. **16**, when the eccentric angle ϕ is 320°, the rotational speed of the driven cam lobe **50***a* decreases in the substantially entire lift zone including a predetermined section (the ramp section at the closing side) in the vicinity of the closing timing of the intake valve **30** as compared with the case in which the eccentric angle ϕ is 0°. In the present embodiment, 320° is used as the eccentric angle ϕ in the above described second high engine speed time. As the time of the second high engine speed higher than the second predetermined engine speed NE2, the time is applicable, in which the engine speed is unintentionally increased more than the normal use range due to an operation error of a transmission in the case of the vehicle equipped with the internal combustion engine **10** including a manual transmission, for example.

When the high engine speed which is not assumed to be used originally is used as in the above described second high engine speed time, the rotational speed of the driven cam lobe **50***a* is high. Therefore, keeping normal mobility of the variable intake valve operating apparatus **34** becomes difficult. More specifically, in such a case, occurrence of valve jump and valve bounce is feared. When such valve bounce occurs, occurrence of failures such as breakage of the component of the valve train and falling-off of the rocker arm **82** is feared.

In the present embodiment, when the engine speed is increased more than the second predetermined engine speed NE2 which is set in advance as the threshold value of the engine speed capable of securing normal mobility, the lift

curve (ϕ=320°) shown in FIG. **16**(B) is selected. Thereby, the rotational speed of the driven cam lobe **50***a* in the vicinity of the closing timing (ramp section) can be decreased. As a result, the lift acceleration in the vicinity of the actual closing timing of the intake valve **30** (acceleration of the intake valve **30** at the time of seating) can be decreased.

According to the above described control, even in the case of the situation exceeding the second predetermined engine speed NE2 due to some reason including the above described irregular case, valve bounce and the like can be avoided. Therefore, reliability of the variable intake valve operating apparatus **34** (similar to the case of being applied to the variable exhaust valve operating apparatus **36**) can be enhanced. Further, in the example shown in FIG. **16**(B), the predetermined section (ramp section) in the vicinity of the opening timing is also included in the deceleration zone, and therefore, occurrence of valve jump and the like in the opening side also can be effectively prevented. As the degree of influence on valve bounce, the degree of influence is larger in the section in the vicinity of the closing timing than in the section in the vicinity of the opening timing, and therefore, a higher priority is given to effective deceleration in the predetermined section in the vicinity of the closing timing. Further, when the eccentric angle ϕ is 320°, the base lift zone after the intake valve **30** is closed becomes the acceleration zone as shown in FIG. **16**(B), but the load from the driven cam lobe **50***a* side does not act on the intake valve **30**, and therefore, the failures as described above do not occur.

Incidentally, in embodiment 1 described above, as the eccentric angle ϕ which is the control amount of the actuator **76**, 0° (equal speed state) and 180° are used, whereby the two lift curves of the intake valve **30** with the same operation angle and different lift peak timings can be obtained. However, in the present invention, the first and the second lift curves in which the same or the substantially same operation angle value is obtained, and the timings in which the lift amounts show the peaks are different from each other are not limited to the lift curves obtained by the above described eccentric angles ϕ (0° and 180°). More specifically, citing the case having the setting of the operation angle and the eccentric angle ϕ shown in FIG. **8**(A) as an example, with respect to any operation angle value OAx other than the operation angle OA1 in the predetermined operation angle variable range (from OAmin to OAmax), the values of two eccentric angles ϕx1 and ϕx2 which realize the operation angle value OAx exist. Even though the eccentric angles ϕx1 and ϕx2 are the eccentric angles at the time of realizing the same operation angle value OAx, if the values of the eccentric angles ϕ differ from each other, in what timing of the lift zone the acceleration/deceleration zones arrive differs, and the acceleration and deceleration amounts differ, and therefore, the lift peak timings thereof differ from each other. Accordingly, depending on the characteristic (shape) of the lift curve which is required of the variable valve operating apparatus in accordance with the operation conditions, the eccentric angles ϕx1 and ϕx2 at the time of realizing the same value as an arbitrary operation angle value OAx may be used besides the combination of 0° and 180°. Further, the operation angle value which is used in this case is not limited to strictly the same value, and may be the value which can be regarded as substantially the same.

Further, in the variable valve apparatuses **34** and **36** of embodiment 1 described above, in order to obtain the two lift curves of the intake valve **30** with the same operation angle and different lift peak timings, the raceway surface **70***a* the center of which is eccentric with respect to the center of rotation of the control sleeve **70** which is rotationally driven by the actuator **76** is included, and rolling of the control roller **64** of the pantographic link mechanism **68** is guided by the raceway surface **70***a*. However, in the present invention, the configuration of the actuator which moves the raceway surface in the plane direction orthogonal to the axial line of the drive camshaft in order to obtain the first and the second lift curves with the operation angle value being the same or substantially the same is not limited to the above described configuration. More specifically, in what timing of the lift zone of the valve the acceleration/deceleration zones arrive, and what acceleration and deceleration amounts are obtained change depending on in what direction the raceway surface moves in the above described plane. Thus, the actuator in the present invention may be configured, for example, to be able to move the guide member including the raceway surface reciprocally between the control position of the raceway surface for obtaining the first lift curve (for example, the control position of the raceway surface **70***a* shown in FIG. **7**(A)), and the control position of the raceway surface for obtaining the second lift curve (for example, the control position of the raceway surface **70***a* shown in FIG. **7**(C)).

Further, in embodiment 1 described above, there is described the control which is with the intake valve **30** as the main target, and discriminatingly uses, in accordance with the operation conditions of the internal combustion engine **10**, a plurality of lift curves differing in lift peak timing by using the variable intake valve operating apparatus **34**. However, the control in the present invention is not limited to the control with the intake valve **30** as the target, but may be control which is with an exhaust valve as the target and discriminatingly uses, in accordance with the operation conditions of the internal combustion engine, a plurality of lift curves differing in lift peak timing.

Further, in embodiment 1 describe above, the control which changes the eccentric angle ϕ which is the control amount of the actuator **76** in accordance with the operation conditions of the internal combustion engine **10** is described with the engine speed as the main control parameter. However, the parameter for determining the operation conditions of the internal combustion engine in controlling the control amount of the actuator in the present invention is not limited to the engine speed, and may be another control parameter (for example, an engine load).

Further, in embodiment 1 described above, an example of discriminatingly using 180° and 0° as the eccentric angle ϕ under the two kinds of operation conditions differing in engine speed (the above described low engine speed time and the above described first high engine speed time) is described. However, the present invention is not limited to the example described above as long as the first rotation angle and the second rotation angle are discriminatingly used under at least two kinds of operation conditions differing in the engine speed. More specifically, the operation condition in which the first rotation angle or the second rotation angle is used may be under a plurality of operation conditions respectively.

Further, in embodiment 1 described above, the raceway surface **70***a* which is formed as a circumferential surface is included. Thereby, when the control sleeve **70** is moved in such a manner that the center of rotation of the drive camshaft **44** and the center line of the raceway surface **70***a* correspond to each other, the rotational speed of the driven cam lobe **50***a* for an interval in which the drive camshaft **44** makes one rotation can be made an equal speed. However, the raceway surface in the present invention does not always have to be a circumferential surface. For example, the raceway surface which is formed into an elliptical shape may be adopted.

Further, in embodiment 1 described above, the description is made by citing the configuration as an example, in which the relative positional relation between the control sleeve **70** and the drive camshaft **44** is set so that the center of rotation of the drive camshaft **44** is located on the locus of the center point of the raceway surface **70***a*, as seen from the axial direction of the drive camshaft **44**. However, the variable valve operating apparatus of the present invention is not necessarily limited to the one that has the above described setting. More specifically, the variable valve operating apparatus may have the setting of the center of rotation of the drive camshaft **44** deviating from the locus of the center point of the raceway surface **70***a* as seen from the axial direction of the drive camshaft **44**.

Further, in embodiment 1 described above, the control sleeve **70** having the raceway surface **70***a* is included for each cylinder as the guide member, and the control sleeves **70** of the respective cylinders are simultaneously rotationally driven by the motor **80** via the one control shaft **78**. However, the present invention is not limited to the configuration like this, and the control sleeve **70** included for each cylinder as the guide member may be rotationally driven for each cylinder with the individually included electric motor, for example.

In embodiment 1 described above, the description is made with the configuration cited as an example, in which the operation force of the driven cam lobe **50***a* is transmitted to the intake valve **30** (valve) via the rocker arm **82** having the rocker roller **82***a*. However, the variable valve operating apparatus in the present invention is not limited to the one having the configuration as described above, and may be the one that is configured so that the driven cam lobe directly drives the valve via a valve lifter, for example.

Further, in embodiment 1 described above, the cam piece **50** integrally having the two driven cam lobes **50***a* is included at the drive camshaft **44** for each cylinder in the rotatable mode. However, the present invention is not limited to the configuration like this, and may be the one in which individual driven cam lobes are rotatably supported by the drive camshaft individually, for example. The present invention may be the one that includes, for example, the link mechanism such as the link mechanism **68**, the guide member having the raceway surface such as the raceway surface **70***a*, and the actuator such as the actuator **76**, for each of the individual driven cam lobes.

Further, in embodiment 1 described above, the link mechanism **68** which is a four joint link connected in a pantograph shape (rhombic shape) by use of the drive arm part **52***a* and the driven arm part **50***b* having the axis center of the drive camshaft **44** as the common center of rotation, and the drive link **56** and the driven link **60** (in other words, used at the angle side in which the above described rotation angle θ is less than 180°). However, the link mechanism in the present invention is not necessarily limited to the one having the configuration like this, and may be the one that is a four joint link which is used at the angle side in which the above described rotation angle θ is larger than 180°, for example.

In embodiment 1 described above, the control sleeve **70** corresponds to "guide member" in the present invention, the control roller side rotary shaft **62** and the control roller **64** correspond to "contact member" in the present invention, and the link plate **66** and the holding roller **72** correspond to "contact maintaining means" in the present invention. Further, "control means" in the present invention is realized by the ECU **40** executing a series of processes of the routine shown in the above described FIG. **12**.

Further, in embodiment 1 described above, 180° and 0° of the eccentric angles φ respectively correspond to "first rotation angle" and "second rotation angle" in the present invention. Further, the lift curve at the time when the eccentric angle φ is 180° (the above described low engine speed time) and the lift curve at the time when the eccentric angle φ is 0° (the above described first high engine speed time) respectively correspond to "first lift curve" and "second lift curve" in the present invention.

DESCRIPTION OF SYMBOLS

10 internal combustion engine
12 piston
14 combustion chamber
16 intake passage
18 exhaust passage
20 air flow meter
22 turbo supercharger
24 throttle valve
26 fuel injection valve
28 ignition plug
30 intake valve
32 exhaust valve
34 variable intake valve operating apparatus
36 variable exhaust valve operating apparatus
38 crank angle sensor
40 ECU (Electronic Control Unit)
42 accelerator position sensor
44 drive camshaft
46 timing pulley
48 variable valve timing (VVT) mechanism
50 cam piece
50*a* driven cam lobe
50*a***1** base circle part of driven cam lobe
50*a***2** nose part of driven cam lobe
50*b* driven arm part of driven cam lobe
52 drive arm
52*a* drive arm part of drive arm
54 camshaft side rotary shaft
56 drive link
58 cam lobe side rotary shaft
60 driven link
62 control roller side rotary shaft
64 control roller
66 link plate
68 link mechanism
70 control sleeve (guide member)
70*a* raceway surface of control sleeve
70*b* gear of control sleeve
72 holding roller
74 holding rotary shaft
76 actuator
78 control shaft
78*a*, **78***b* gear of control shaft
80 electric motor
80*a* output shaft of electric motor
80*b* gear at electric motor side
82 rocker arm
82*a* rocker roller
84 hydraulic lash adjuster
86 valve spring

The invention claimed is:

1. A variable valve operating apparatus for an internal combustion engine, comprising:

a drive camshaft that is rotationally driven by a rotational force of a crankshaft;

a driven cam lobe that is concentric with the drive camshaft, and is rotatably supported by the drive camshaft;

a guide member that has a raceway surface formed so as to surround the drive camshaft;

a link mechanism that is connected to each of the drive camshaft and the driven cam lobe, has a contact member which is in contact with the raceway surface, and changes a rotation angle of the driven cam lobe with respect to the drive camshaft in association with a positional change of the contact member with respect to a center of rotation of the drive camshaft;

a contact maintaining device that maintains contact between the contact member that revolves around the drive camshaft and the raceway surface, while the drive camshaft makes one rotation;

an actuator that moves the raceway surface in a plane direction orthogonal to an axial line of the drive camshaft; and

a controller that is configured to control a control amount of the actuator to change a moving amount of the raceway surface in the plane direction, in accordance with an operation condition of the internal combustion engine,

wherein, in a state in which a center of the raceway surface is eccentric from a center of rotation of the drive camshaft, a distance of the contact member with respect to the center of rotation of the drive camshaft is changed while the drive camshaft makes one rotation.

2. The variable valve operating apparatus for an internal combustion engine according to claim **1**,

wherein the actuator rotationally drives the guide member,

wherein the raceway surface is a circumferential surface, and is included by the guide member in a state in which the center of the raceway surface is eccentric with respect to a center of rotation of the guide member, and

wherein the control amount of the actuator by the controller is a rotation angle of the guide member.

3. The variable valve operating apparatus for an internal combustion engine according to claim **2**,

wherein the controller includes at least a first rotation angle and a second rotation angle as target values of the rotation angle of the guide member,

wherein the first rotation angle is a rotation angle of the guide member at a time when a predetermined operation angle value for an operation angle of a valve which is driven by the driven cam lobe is obtained,

wherein the second rotation angle is a rotation angle of the guide member at a time when a same or substantially the same operation angle value as the operation angle value which is obtained at a time of control to the first rotation angle is obtained, and when a second lift curve with timing in which a lift amount shows a peak differing as compared with a first lift curve of the valve which is obtained at the time of the control to the first rotation angle is obtained, and

wherein the controller discriminatingly uses the first rotation angle and the second rotation angle under at least two kinds of operation conditions which differ in engine speed from each other.

4. The variable valve operating apparatus for an internal combustion engine according to claim **3**,

wherein the valve is an intake valve,

wherein the second lift curve is set so that timing in which the lift amount shows the peak is delayed as compared with the first lift curve, and

wherein the controller uses the second rotation angle in an operation condition with a higher engine speed than an operation condition for which the first rotation angle is used.

5. The variable valve operating apparatus for an internal combustion engine according to claim **4**,

wherein the second lift curve is set so that a lift amount in a predetermined section in a vicinity of a closing timing of the valve becomes small as compared with the first lift curve.

6. The variable valve operating apparatus for an internal combustion engine according to claim **1**,

wherein a valve which is driven by the driven cam lobe is an intake valve, and

wherein when an acceleration request is issued at a time of a low engine speed, the controller controls the control amount of the actuator so that a relative rotational speed of the driven cam lobe with respect to the drive camshaft increases in a predetermined section in a vicinity of an opening timing of the intake valve.

7. The variable valve operating apparatus for an internal combustion engine according to claim **1**,

wherein when an engine speed is higher than a predetermined speed, the controller controls the control amount of the actuator so that a relative rotational speed of the driven cam lobe with respect to the drive camshaft decreases in a predetermined section in a vicinity of a closing timing of a valve which is driven by the driven cam lobe.

8. A variable valve operating apparatus for an internal combustion engine, comprising:

a drive camshaft that is rotationally driven by a rotational force of a crankshaft;

a driven cam lobe that is concentric with the drive camshaft, and is rotatably supported by the drive camshaft;

a guide member that has a raceway surface formed so as to surround the drive camshaft;

a link mechanism that is connected to each of the drive camshaft and the driven cam lobe, has a contact member which is in contact with the raceway surface, and changes a rotation angle of the driven cam lobe with respect to the drive camshaft in association with a positional change of the contact member with respect to a center of rotation of the drive camshaft;

contact maintaining means that maintains contact between the contact member that revolves around the drive camshaft and the raceway surface, while the drive camshaft makes one rotation;

an actuator that moves the raceway surface in a plane direction orthogonal to an axial line of the drive camshaft; and

control means that controls a control amount of the actuator to change a moving amount of the raceway surface in the plane direction, in accordance with an operation condition of the internal combustion engine,

wherein, in a state in which a center of the raceway surface is eccentric from a center of rotation of the drive camshaft, a distance of the contact member with respect to the center of rotation of the drive camshaft is changed while the drive camshaft makes one rotation.

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